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THE ENERGETIC AND EXERGETIC EVALUATION OF THE EXHAUST GASES ON THE EXAMPLE OF THE SELECTED MARINE DIESEL ENGINES

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Abstract

The tendency to improve the efficiency of the utilisation of the primary energy on board the ships, as well as the anticipated necessity of the obtainment of the Energy Efficiency Design Index (EEDI) in accordance with the requirements of the International Maritime Organisation (IMO) induce to apply the power system solutions which are among others characterised by low CO_2 emission.

The way to improve the ship's power plant energetic efficiency, thus lowering CO_2 emission on motor ships, is to apply continuously more developed recovery systems of the waste energy whose basic sources are the piston Diesel engines consisting the ship's main propulsion. The most significant component of the waste energy is the energy of the exhaust gases. In order to efficiently utilise this energy not only its quantity, but also its quality should be evaluated.

The article presents the examples of the energetic evaluation of the energy of the flue gases of the selected ship's main propulsion Diesel engines and the examples of the exergetic evaluation of this energy. The commonly applied in design practice energetic analysis in principle allows to determine the fundamental possibilities of the utilisation of the exhaust gas energy in the systems with waste-heat boilers. The exergetic evaluation which includes the temperature and pressure parts of the physical exergy offers a possibility to analyse more complex systems where so called exhaust gas reverse turbine or waste-heat Diesel turbogenerator are used. The latter solutions including the part where the energy carrier is steam are becoming nowadays the object of interest of the Diesel engine manufacturers and ship owners, as well as the subjects of research works and study projects concerning the efficient and low-emission ship's power plants.

Key words: *ship's power plants, utilisation of waste energy, exhaust gas, exergetic analysis*

1. Introduction

The tendency to improve the efficiency of the utilisation of the primary energy on ships, as well as the anticipated necessity to obtain the Energy Efficiency Design Index (EEDI) in compliance with the requirements of the International Maritime Organisation (IMO)) induce to apply power system solutions of high general efficiency which owing to this are also among others characterised by a relatively low CO_2 emission [2]. Other IMO documents concerning the necessity of reduction of CO_2 emission are Energy Efficiency Operational Indicator (EEOI) [3] and Ship Energy Efficiency Management Plan (SEEMP) [1]. The analysis of each of the aforesaid documents indicates that a method to improve the ship's power plant energy efficiency, thus limiting CO_2 emission on motor ships is the application of the continuously developed recovery systems of waste energy whose basic sources are the Diesel piston engines consisting ship's main propulsion.

The development of the ship's main and auxiliary propulsion system engine construction makes their efficiency visibly increase. However, it becomes more difficult to recover the waste energy due to its lowering quality. Thus is becomes very important to utilise this energy which ensures the improvement of the general ship's operation cost balance. The appropriate evaluation of the waste energy sources consists the basis for the choice of the manner of its utilisation, and then subsequently elaborating the ship's power system design. This evaluation should cover not only the parameters of the waste energy carriers corresponding to the nominal engine load, but also should include their change resulting from the change in engine load during ship's service.

2. Balance of the Energy Flux of the Ship's Diesel Piston Engines

The modern ship's Diesel engines effectively utilise approximately 50 % of the energy contained within the combusted fuel. Despite the high efficiency of these engines there still remains a significant part of unused fuel energy as so called "waste" energy. Thus apart from the mechanical energy there is also discharged to the environment the energy contained in: exhaust gas, cylinder cooling water, charging air cooling water, piston coolants, lubricating oil, fuel valve coolants as well as in the convection processes and heat transfer to the engine ambient air. A part of the waste energy, owing to its relatively high quality factor, can be further utilised in economically viable manner [5, 10].

There are distinguished the physical waste energy and the chemical waste energy related to the operation of ship's Diesel piston engines. The physical waste energy appears in the temperature part, the form which results from the deviation of the temperature of the energy waste carrier from the ambient temperature and the pressure part resulting from the increased pressure in relation to the ambient prevailing pressure. The chemical waste energy is the result of the difference in the chemical composition of the waste substance which is exhaust gas in relation to the commonly present components of the environment [8, 9, 10]. This part of the waste energy is, however, generally utilised.

In order to assess the waste energy resources so called engine heat balance is done. A supplementary information for this energy form is the knowledge of the temperature, pressure and properties of its carriers. The analysis of heat balance of the commonly used as the ship's main propulsion slow speed piston Diesel engines of MAN Diesel and Wärtsilä make indicates that the heat utilised efficiently consists (chiefly subject to the engine rating or power output) 47.1 ÷ 50.9 % of the energy contained in the combusted fuel. The heat transferred in the exhaust gas consists respectively 21.5 ÷ 25.5 %, in charging air cooling water – 15.6 ÷ 19.5 %, in cylinder cooling water 6.5 ÷ 10.5 % whereas the heat contained in luboil – 3.8 ÷ 6.3 %. The exhaust gas temperature in ISO Standard conditions and at the maximum continuous rating (MCR) varies within 508 ÷ 548 K, charging air cooling water temperature 318 ÷ 331 K, cylinder cooling water outlet temperature 353 ÷ 363 K, oil temperature at the engine outlet 323 ÷ 348 K [6]. It should be noted that the relatively high value of heat amount does not always correspond to the high temperature of the heat carriers. This is for instance the case of the heat contained in charging air cooling temperature. The precise analysis of the parameters referred to above frequently causes leaving aside the sources advantageous in terms of the amount of heat carried and paying more attention to others, characterised for instance by a bigger capacity to perform a work.

Considering the fact that the waste energy source of the biggest energetic potential in motor power plants (and also turbo-Diesel power plants) are the engine exhaust gases, the further part of this article shall be devoted chiefly to the example of the evaluation of the energy contained therein.

3. Exhaust Gas Energy and Exergy

The basis for the evaluation of engine exhaust gas energy amount is the knowledge of the exhaust gas temperature and the specific exhaust gas amount. The exhaust gas temperature after the turbochargers in the design conditions, similarly as the specific exhaust gas amount, is a function of many variables. In order to determine it, one should determine its value corresponding to the maximum continuous rating (MCR) of a given engine in the ISO Standard conditions for a given turbocharger type and pressure value after the turbocharger corresponding to MCR. The calculated exhaust gas temperature is then revised subject to: location in the field of the contractual parameters, so called Contract Maximum Continuous Rating (CMCR), also referred to as Specified Maximum Continuous Rating (SMCR) and the location of the optimised engine operation point. Attention should also paid to the ambient conditions (air temperature at compressor inlet, ambient pressure, water temperature at the charging air inlet), exhaust gas pressure after turbocharger corresponding to the location of the optimised engine operation point in relation to the parameters corresponding to MCR and chiefly engine load resulting from its assumed power output and crankshaft rpm. It also depends on whether the fixed pitch propeller (FPP) or controllable pitch propeller (CPP) is applied as the ship's main propulsion. The exhaust gas temperature will be the lower the lower rpm and power output is determined by CMCR point in relation to the parameters corresponding to MCR point and engine load will be lower in the result of the reduced power rating. The decrease in exhaust gas temperature is also caused by the increase in the barometric pressure. The increase of the air temperature at the turbocharger inlet and the increase of the water temperature before charging air cooler and increase of pressure at the engine turbocharger turbine outlet cause the exhaust gas temperature to grow. The illustration of the discussed changes are the data in table 1 specific for MAN Diesel low-speed engines [4].

Tab. 1. Correction of exhaust gas data to the ambient conditions and exhaust gas back pressure [4]

Parameter	**Change**	**Change of exhaust gas temperature**	**Change of exhaust gas amount**
Blower inlet temperature	+ 10°C	+16.0°C	-4.1 %
Blower inlet pressure (barometric pressure)	+ 10 mbar	-0.1°C	+0.3 %
Change in air coolant temperature (seawater temperature)	+ 10°C	+1.0°C	+1.9 %
Exhaust gas back pressure at the specified MCR point	+ 100 mm WC	+5.0°C	-1.1 %

The examples of the correction of the temperature value DTs and specific gas amount Dms resulting from the change of the MAN L70 MC-C power output are demonstrated in figures 1 and 2 [4].

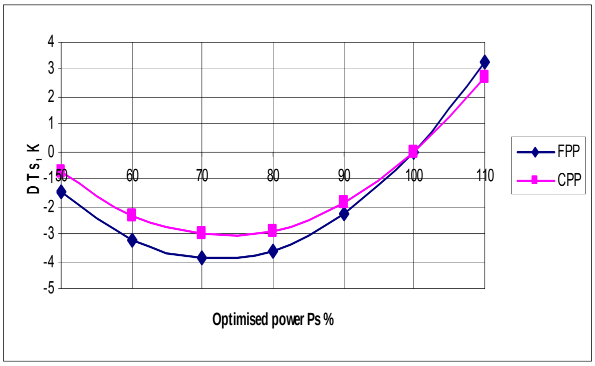


Fig.1. Change of exhaust gas temperature, DTs, at part load [4]

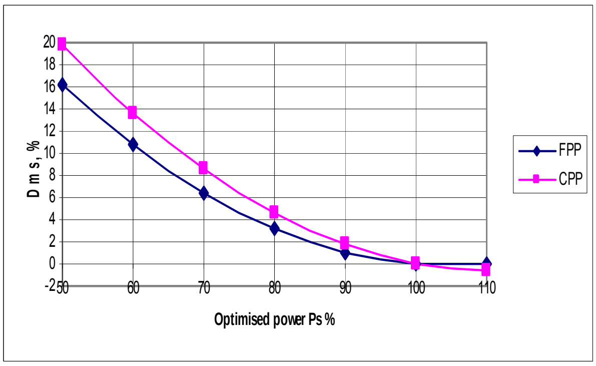
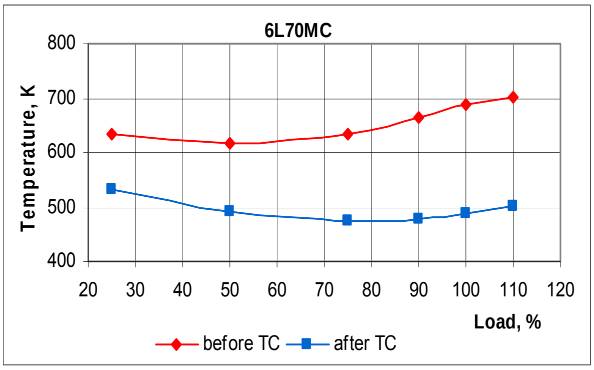


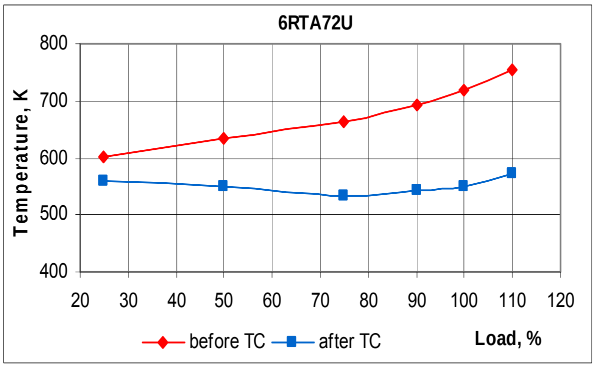
Fig.2. Change of specific exhaust gas amount, Dms, in % at part load [4]

The courses of the changes in the temperatures of exhaust gas before and after turbocharger in the function of the loads of the selected low-speed and medium-speed engines are shown in figures 3. These have been elaborated thanks to Messrs Cegielski in Poznań having made available the tests and trials results on their factory test bed. As an example there have been quoted the values of exhaust gas parameters of 6L70 MC-C engine of MAN Diesel make of 17,200 kW power at 108 rpm, provided with the 1xMAN B&W NA 70/T09 turbocharger, running on ISO-F-DMB fuel; 6RTA72U Wärtsilä make, of 17,940 kW power and 97 rpm, provided with 2xABB TPL77-B11 turbochargers, running on ISO-F-DMB fuel and 8L32/40 engine of Man Diesel make of 3,840 kW and 750 rpm provided with 1xMAN NR34/SO30 turbocharger, running on MDF-D.O. LII fuel.

a)



b)



c)

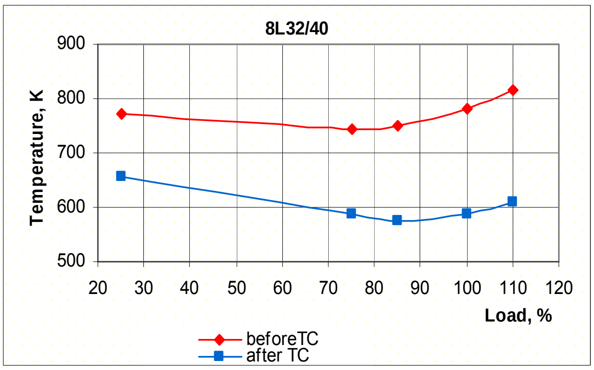


Fig 3. The course of the exhaust gas temperature changes before and after turbochargers a) 6L70MC-C engine; b) 6RTA52U engine; c) 8L32/40 engine

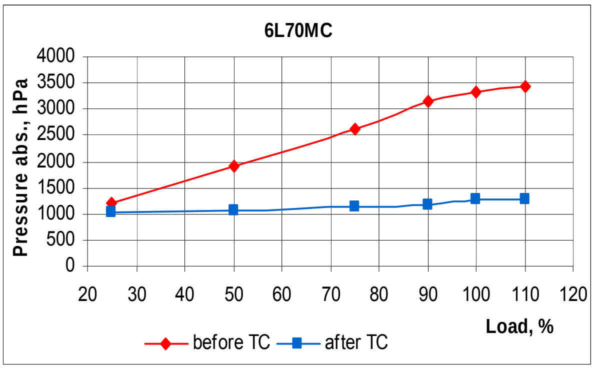
The data shown in figure 1 indicate a strong coincidence with the results obtained during the tests of the engine on the factory test bed (figure 3a), in particular as far as the quality of the exhaust gas temperature changes after turbocharger is concerned. It should be noted that within the area of the maximum engine efficiencies the temperature of exhaust gas reaches the lowest values for the obvious reasons. It is also worth emphasising that the exhaust gas temperatures before turbochargers are higher than the average temperature of the exhaust gas leaving the cylinders due to the phenomenon of the swelling occurring during the exhaust gas flow from the cylinders to the big volume outlet [7].

In cases of the recovery systems where steam is generated or special oils are heated, it is important to know mainly the course of the temperature changes of the exhaust gas leaving the turbocharger. Whereas, if one considers the variant with the internal combustion waste heat turbine running in the system Power Take In (PTI) or Power Take Off (PTO), it is important to know not only the temperature but also the pressure of exhaust gas before turbine (in exhaust gas tank).

The inclusion of this parameter is of particular significance while evaluating the exhaust gas capacity for performing a work in the internal combustion waste heat turbine. The exhaust gas pressure before the turbine allows to determine the possible expansion in the turbine. The courses of the exhaust gas pressure changes before and after turbochargers are demonstrated on figures 4.

As shown by the data quoted in figures 4, this expansion at the nominal load of the engines in question, both low-speed as the medium-speed, slightly exceeds the value 3.

a)



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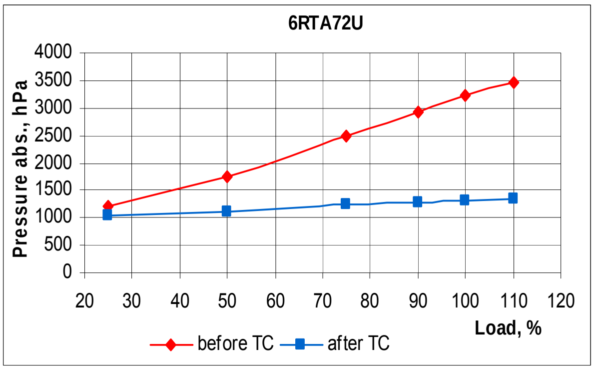

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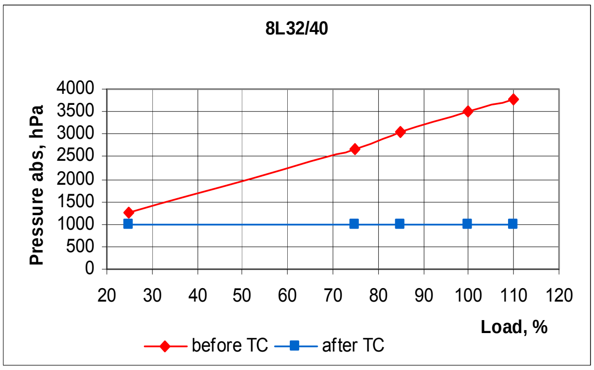
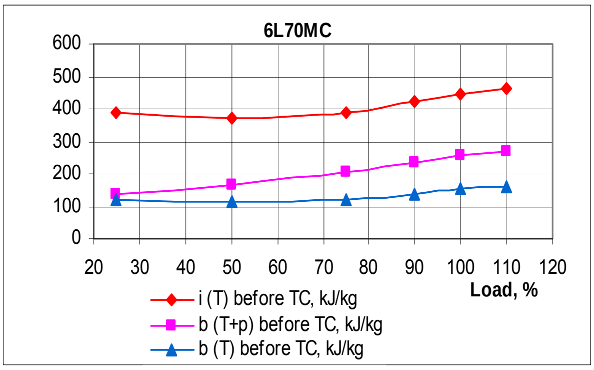


Fig 4. The course of the changes of the absolute pressure of exhaust gas before and after turbochargers
a) 6L70MC-C engine; b) 6RTA72U engine
c) 8L32/40 engine

While analysing the complex systems of the recovery of the waste energy, particularly those using IC turbines, it is useful to consider the exergy concept for the evaluation of the waste energy quality. The knowledge of exergy of the carriers under consideration allows to evaluate more closely the quality [8, 9, 10]. And while designing the ship's systems of waste energy recovery it is sufficient to know the physical exergy. The specific physical exergy of the exhaust gas "*b*", comprising the temperature and pressure parts can be determined according to the method presented in [6].

The data presented in figures 3 and 4 allowed to determine the value of the specific exergy of the exhaust gas before and after engine turbochargers. The results of the calculations have been presented in the graphic form in the figures 5 and 6, narrowing the area of the example to the low-speed engines. For the comparative purposes there have been demonstrated also the courses in changes of the specific enthalpy "*i*" of exhaust gas in the function of engine load.

a)



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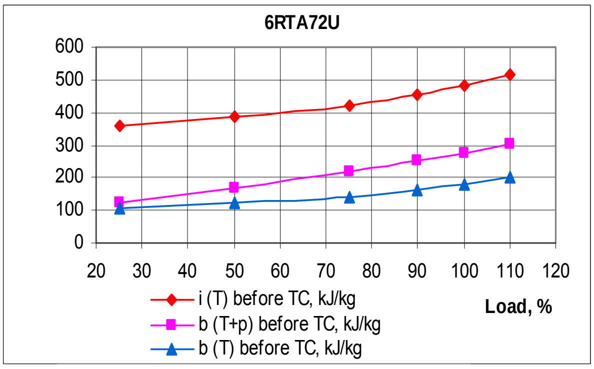

Fig 5. The course of the changes in the specific enthalpy and (T), specific exergy including the temperature and the pressure part b (T+p) part and the specific exergy including the temperature part b (T) of the exhaust gas before turbochargers
a) 6L70MC-C engine; b) 6RTA72U engine

a) b)

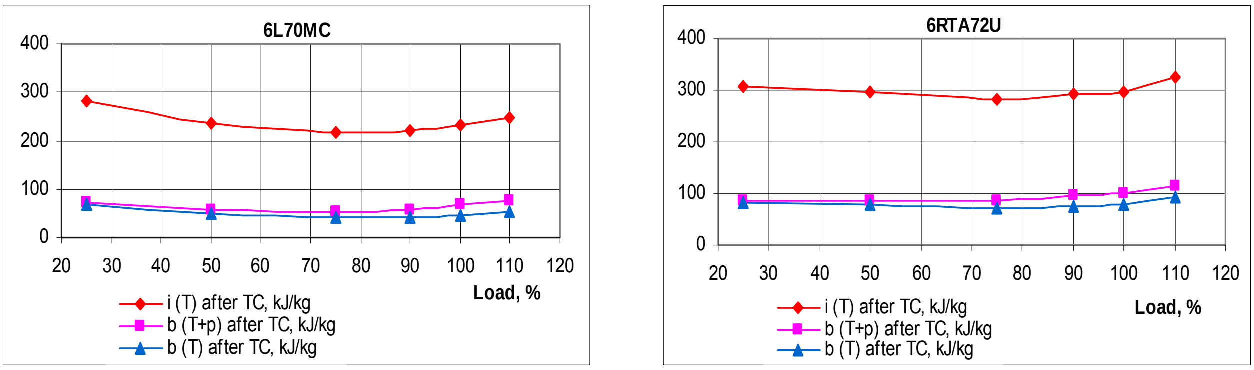

Fig 6. The course of the changes in the specific enthalpy and (T), specific exergy including the temperature and the pressure b (T+p) part and the specific exergy including the temperature part b (T) of the exhaust gas after turbochargers
a) 6L70MC-C engine; b) 6RTA72U engine

With the engine loads in order of 90% the exhaust gas specific exergies in case of 6RTA72U are slightly higher in comparison with the specific exergies of the exhaust gas leaving 6L70MC-C engine. This practically offers in this manner the similar possibilities of the utilisation of this form of energy.

The usability of the resources of the low temperature waste energy can also be evaluated by means of so called energy quality index [10]. This index means the maximum share of the work possible to attain for the assumed ambient conditions, in relation to the heat amount contained in the waste energy carrier.

$$\psi = \frac{T_m - T_o}{T_m} \tag{1}$$

where T_m – average thermodynamic temperature of the waste energy carrier, K,
T_o – ambient temperature, K.

The temperature part of exergy b_T can be expressed by means of Carnote's factor:

$$b_T = \Delta i_p \frac{T_m - T_o}{T_m} \tag{2}$$

where Δi_p – isobaric enthalpy increase of a substance under pressure p within the ambient temperature and the temperature of substance in question.

Thus the energy quality index can be determined as:

$$\psi = \frac{b_T}{\Delta i_p} \tag{3}$$

Using the model presented above there have been calculated energy quality indices including only temperature part and those including both temperature and pressure parts of the exhaust gas both before and after turbochargers.

The figures 7 and 8, by way of example, present the quality indices of the exhaust gas of 6L70MC-C and 6RTA72U engines calculated according to equation 3, both before and after turbochargers. Under 90% load index including the temperature part ψ (T) of exhaust gas of MAN Diesel engine amounts to 32.9 %, after turbocharger 19.6 %. If one considers the pressure part these indices ψ (T+p), shall amount respectively over 56.2 % and 26.2 %. In case of Wärtsilä engine, before turbochargers ψ (T) = 35.6 %, ψ (T+p) = 55.0 %, whereas after the turbochargers ψ (T) = 26.1 %, and ψ (T+p) = 33.0 % respectively.

a) b)

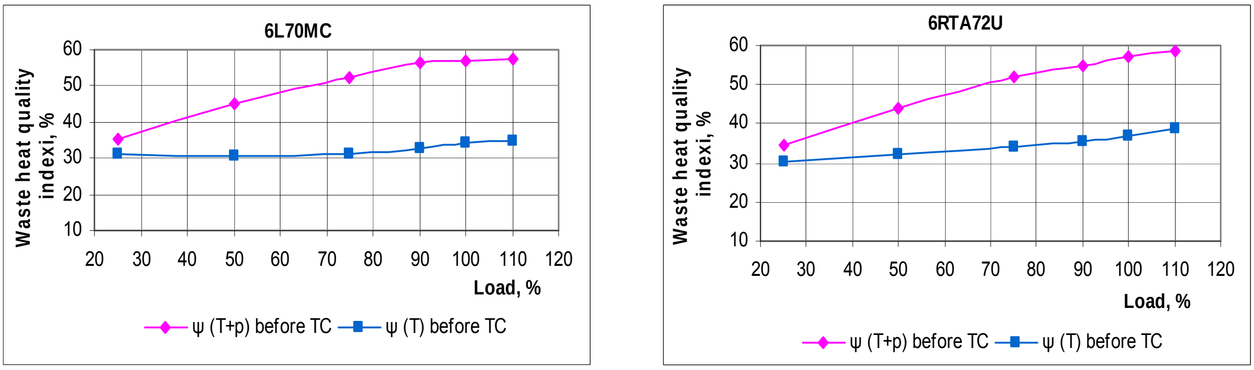

Fig 7. The course of the changes of the heat energy quality index including both the temperature and pressure parts ψ (T+p) as well as same including only temperature part ψ (T) of the exhaust gas before turbochargers a) 6L70MC-C engine; b) 6RTA52U engine

a) b)

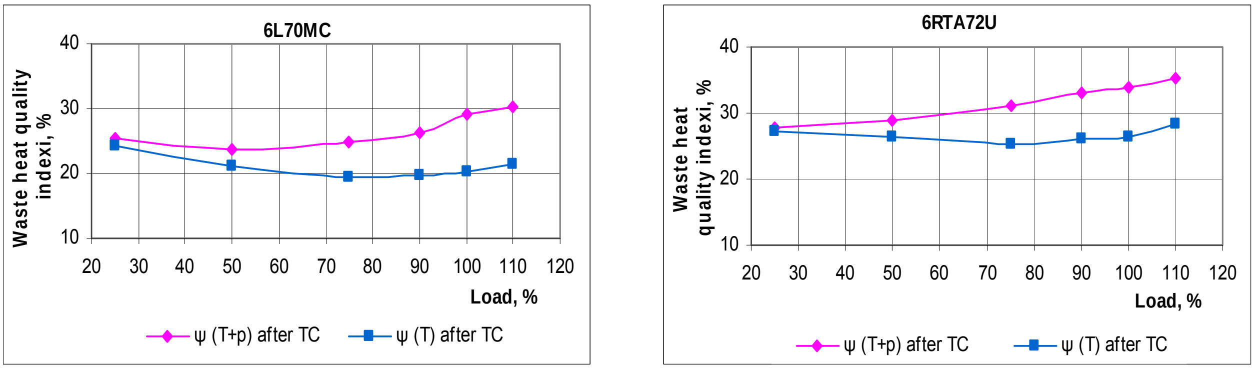

Fig 8. The course of the changes of the heat energy quality index including both the temperature and pressure parts ψ (T+p) as well as same including only temperature part ψ (T) of the exhaust gas after turbochargers of a) 6L70MC-C engine; b) 6RTA52U engine

4. Conclusions

In the design practice, in the ship's recovery systems chiefly the energy/heat contained in exhaust gas and cylinder cooling water is utilised, less frequently that in the engine charging air and only occasionally that contained in the lubricating oil. The biggest part of waste energy/heat is contained in the engine exhaust gas.

The knowledge of the exhaust gas parameters allows to take a reasonable decision concerning the choice of one of many possible variants of the recovery system solutions. While designing such systems one should consider not only the exhaust gas parameters corresponding to the calculated continuous service rating at which the engine rating is usually 85 to 90 (92) % of the nominal continuous rating (CMCR), and the rpm consist respectively 95 to 97 % of the nominal rpm value, but also the exhaust gas parameters corresponding to other partial ratings. The available amount of waste energy/heat contained in exhaust gas is decreasing on account of their decreasing specific amount despite the increase of their specific exergy within certain engine loads, At that time the total ship's need of heat decreases but slightly.

The evaluation of the sources of waste energy made on the basis of the engine heat balance, even in connection with the information on the temperature of the heat carriers fails to provide the explicit and clear information on its usability. An important component of the exhaust gas exergy is its pressure part. Information on this exergy part is significantly meaningful when designing the recovery systems with internal combustion turbines.

The application of the exergetic analysis in connection with the energetic analysis for the evaluation of the waste energy allows to obtain more comprehensive information on the quality and maximum possibilities of the utilisation of this energy.

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