Thermodynamic Analysis and Comparison of Two Marine Steam Propulsion Turbines

Termodinamička analiza i usporedba dviju propulzijskih parnih turbina

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DOI 10.17818/NM/2023/2.2
UDK 621.125: 629.5.03
Original scientific paper / *Izvorni znanstveni rad*
Paper received / *Rukopis primljen:* 28. 2. 2023.
Paper accepted / *Rukopis prihvaćen:* 20. 3. 2023.

Abstract

This paper presents thermodynamic (energy and exergy) analysis and comparison of two different marine propulsion steam turbines based on their operating parameters from exploitation. The first turbine did not possess steam reheating and had only two cylinders (high-pressure and low-pressure cylinders), while the second turbine possesses steam reheating and has one additional cylinder (intermediate-pressure cylinder). In the literature at the moment, there cannot be found a direct and exact comparison of these two marine steam turbines and their cylinders based on real exploitation conditions. Along with energy and exergy analyses, the research it is investigated the sensitivity of exergy parameters to the ambient temperature change for both turbines and each cylinder. It is also presented the influence of the steam reheating process on the energy and exergy efficiency of the entire power plant. For both observed turbines and their cylinders it is valid that relative losses and efficiencies (both energy and exergy) are reverse proportional. The operation of an intermediate pressure cylinder from a steam turbine with reheating is the closest to optimal. Due to the different origins of losses considered in energy and exergy analyses, each analysis detects different turbine cylinders as the most problematic ones. The steam reheating process decreases losses and increases efficiencies (both energy of each turbine cylinder and the whole turbine. The whole turbine with reheating has an energy efficiency equal to 81.46% and an exergy efficiency equal to 86.48%, while the whole turbine without reheating has energy and exergy efficiencies equal to 76.47% and 80.94%, respectively. Exergy parameters of a steam turbine without reheating as well as its cylinders are much more influenced by the ambient temperature change in comparison to the steam turbine with reheating and its cylinders. The steam reheating process will increase the efficiency of the whole power plant in real exploitation conditions between 10% and 12%.

KEY WORDS

marine propulsion
steam turbines
energy analysis
exergy analysis
ambient temperature variation

Sažetak

Ovaj rad predstavlja termodinamičku energijsku i eksergijsku analizu te usporedbu dviju propulzijskih parnih turbina na temelju njihovih radnih parametara iz eksploatacije. Prva turbina ne posjeduje pregrijavanje pare i ima samo dva kućišta (visokotlačno i niskotlačno kućište), dok druga turbina posjeduje pregrijavanje pare i ima jedno dodatno kućište (srednjetlačno kućište). U literaturi se trenutačno ne može naći izravna i egzaktna usporedba ovih dviju brodskih parnih turbina i njihovih kućišta bazirana na stvarnim eksploatacijskim parametrima. Uz analize energije i eksergije, u istraživanju ispituje se osjetljivost parametara eksergije u odnosu na izmjenu temperature ambijenta za obje turbine i svako kućište. Također je prikazan utjecaj procesa pregrijavanja pare na energijsku i eksergijsku iskoristivost čitavoga postrojenja. Za obje promatrane turbine i njihova kućišta vrijedi da su relativni gubici i iskoristivost (energije i eksergije) obrnuto proporcionalni. Djelovanje srednjetlačnog kućišta parne turbine sa pregrijavanjem pare najbliže je optimalnom. Zbog različitih izvora gubitaka pri energijskoj i eksergijskoj analizi, svaka analiza detektira različita kućišta turbina kao najproblematičnija. Proces pregrijavanja pare smanjuje gubitke i povećava iskoristivost (energije i eksergije) svakoga kućišta i cijele turbine. Cijela turbina s pregrijavanjem pare ima energetsku iskoristivost koja je jednaka 81,46% i eksergijsku iskoristivost koja je jednaka 86,48%, dok cijela turbina bez pregrijavanja pare ima energijsku i eksergijsku iskoristivost koja je jednaka 76,47%, odnosno 80,94%. Eksergijski parametri parne turbine bez pregrijavanja pare, kao i njezina kućišta pod većim su utjecajem promjena ambijentalne temperature u usporedbi s parnom turbinom sa pregrijavanjem pare i njezinim kućištima. Proces pregrijavanja pare povećat će iskoristivost cijeloga pogonskog postrojenja u stvarnim eksploatacijskim uvjetima u iznosu od 10% do 12%.

KLJUČNE RIJEČI

propulzija broda
parne turbine
energijska analiza
eksergijska analiza
varijacija ambijentalne temperature

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1. INTRODUCTION / *Uvod*

In the shipping sector nowadays diesel engines prevail as the dominant mechanical power producers [1-4]. On ships, conventional diesel engines can be used as the main propulsion devices (usually slow-speed two-stroke diesel engines) [5-7] or can be used as auxiliary mechanical power producers (usually medium-speed four-stroke diesel engines) [8, 9]. The new technologies and improvements, especially in the field of harmful emissions reduction due to stringent legislation affects also marine diesel engines [10-12]. The legislation related to harmful emissions [13, 14] was the reason that conventional diesel engines are being more and more replaced by dual-fuel engines whose operation (especially in the gas operating mode) notably reduces emissions [15-17]. Also, the process of dual-fuel engines is still improving each day intending to obtain optimal operation [18-20].

Instead of a conventional diesel or dual-fuel engines in the shipping industry, especially on ships where a high amount of mechanical power is required, can be used steam power plants and steam turbines [21, 22]. In steam power plants onboard ships, steam turbines are traditionally used as the main turbines (for the propulsion propeller drive) [23, 24] and as auxiliary turbines (for the electrical generators or pump drive) [25, 26]. Auxiliary steam turbines are usually low-power turbines that have only one cylinder [27, 28] and can be composed of only one (Curtis) stage [29].

Main steam turbines used for the propulsion propeller drive are usually composed of two or three cylinders [30]. Two-cylinder propulsion turbines did not possess steam reheating (older versions), while three-cylinder propulsion turbines (newer versions) possess steam reheater between high-pressure and intermediate-pressure cylinders [31, 32]. Both steam turbines have an additional turbine (mounted in the same housing as a low-pressure cylinder) for the astern drive [33]. Marine steam power plants with steam reheating represent the latest achievement in the shipping industry, while in land-based steam power plants, steam reheaters are standard plant elements for many years [34-36]. In the marine sector, the steam reheating process allows a notable increase in steam pressure at the steam generator outlet (in comparison to marine steam power plants which did not possess steam reheating) which brings many benefits as well as some disadvantages such as power plants [37-39].

In the available literature related to marine steam turbines with and without reheating (marine steam turbine with reheating is usually called UST – Ultra Steam Turbine [39]) can be found general guidelines related to the advantages and possible disadvantages which reheating process brings in the steam turbine and entire plant operation. There is missing exact data related to the steam reheating process's influence on the steam turbine, its cylinders, and the entire plant operation. Currently, it is unknown which cylinder operation is the closest to optimal, which efficiencies can be achieved (for the whole turbine and each cylinder), and how the ambient temperature change influences UST operation. Also, the literature offers general recommendation that the steam reheating process can increase overall marine plant efficiency by up to 15% [39], but exact values and comparison with a process without reheating, based on a real operating parameter obtained in exploitation, cannot be found in the literature at the moment.

This research fulfills the literature gap because it offers a direct and exact comparison of two marine steam turbines (with and without the steam reheating process) based on their operating parameters from exploitation. The analyses performed in this research present which cylinder of both observed turbines is the dominant mechanical power producer and which losses are the most influential from the energy and exergy aspect. For the whole turbine and each turbine cylinder (both observed steam turbines) there are presented exact losses and efficiencies which can be achieved in exploitation. It is analyzed which of the two observed marine steam turbines (as well as their cylinders) are more influenced by the ambient temperature change. At the end of this research, it is presented which exact efficiency increase of the whole marine power plant with a steam reheater can be expected during exploitation in comparison to the power plant which did not possess a steam reheater.

2. DESCRIPTION AND OPERATION CHARACTERISTICS OF THE ANALYZED MARINE PROPULSION STEAM TURBINES / *Opis i radne karakteristike analiziranih brodskih propulzijskih parnih turbina*

The analysis and comparison in this paper are performed for two different marine propulsion steam turbines – the first one is a turbine that did not possess a steam reheater, while the second one has a steam reheater. Simplified schemes of the observed marine propulsion steam turbines along with operating points required for their thermodynamic analysis are presented in Figure 1 (a) for a turbine without reheating and in Figure 1 (b) for a turbine with reheating.

Both analyzed marine steam turbines are used for the ship propulsion propeller drive which must be performed by using a gearbox (for both observed turbines). Steam turbines have high rotation speeds and their direct connection with a propulsion propeller (as is the case of slow-speed two-stroke diesel engines) is not possible. The observed steam turbine without reheating operates at the Liquefied Natural Gas (LNG) carrier [31], while the steam turbine with reheating operates at the crude oil carrier [32].

The first observed marine propulsion steam turbine without reheating, Figure 1 (a), has only two cylinders – High-Pressure Cylinder (HPC) and Low-Pressure Cylinder (LPC). The dominant amount of superheated steam produced in the steam generator is delivered to the propulsion turbine (operating point 4, Figure 1 (a)), while the rest of produced superheated steam is delivered to auxiliary marine steam turbines (turbogenerators and the turbine for the main feedwater pump drive – operating point 3, Figure 1 (a)). The steam expands firstly through the HPC (HPC has one extraction for steam delivery to auxiliary ship systems). Between HPC and the LPC is no mounted steam reheater, so the steam, after expansion in HPC is delivered directly to LPC (between HPC and the LPC is mounted another extraction for steam delivery to high-pressure feed water heater and deaerator - operating point 7, Figure 1 (a)). The HPC, also LPC have one extraction for steam delivery to the low-pressure condensate heater and evaporator (operating point 9, Figure 1 (a)). After expansion through LPC, the remaining steam mass flow rate is delivered to the main seawater-cooled condenser for condensation.

The second marine propulsion steam turbine considered in this analysis, Figure 1 (b), along with HPC and the LPC has one additional cylinder - Intermediate Pressure Cylinder (IPC). In this turbine, steam produced in the steam generator is delivered first to the HPC – HPC has two extractions for steam delivery to two high-pressure feedwater heaters. After expansion in HPC, the remaining steam mass flow rate is delivered to the reheater mounted in the steam generator which increases steam temperature before its expansion in IPC (operating points 6 and 7, Figure 1 (b)). In the steam power plant, the steam reheater can be mounted independently of the steam generator [40, 41], but the most common arrangement, not only in marine but also in conventional steam power plants is to place the steam reheater inside the steam generator [42, 43]. After reheating, the steam expands through IPC which has only one extraction for steam delivery to the deaerator. After expansion in IPC, the remaining steam mass flow rate is directly delivered to LPC (operating point 9, Figure 1 (b)) which has two extractions for steam delivery to low-pressure condensate heaters. At the end of expansion in LPC, the remaining steam mass flow rate is delivered to the main condenser for condensation.

The literature [37-39] there can be found various benefits of marine steam power plants with reheating in comparison to marine steam power plants without reheating. Marine steam power plants with reheating have lower specific fuel consumption, higher reliability and safety, lower harmful emissions (especially NO_x and CO_2 emissions), and longer plant life. Also, the steam reheating process in the marine power plant brings much higher steam pressure at the HPC inlet in comparison to the marine power plant which did not possess a reheater (approximately 100 bar in comparison to approximately 60 bar). The high steam pressure of approximately 100 bar at the HPC inlet (plant with reheater) can have some important negative effects on the turbine operation. Such high steam pressure increases axial forces on the turbine rotor which results in more complex axial bearings and an increase in lube oil consumption. High pressure at the HPC inlet also notably increases losses on the first (regulation) turbine stage, so the producers usually recommended expensive 3D blades with optimized angles on that stage [39]. Moreover, high pressure at the HPC inlet increases steam losses through inner and gland seals, so it is recommended to adopt improved sealing techniques at high-pressure seals [39]. Finally, it should be highlighted that, along with all the benefits which the steam reheating process brings to the marine propulsion plant, there exist many challenges which should not be ignored.

As one of the goals of this analysis and comparison was to observe the influence of the steam reheating process on the overall power plant efficiency, it is necessary to know the fuel chemical energy released in a steam generator or at least the amount of energy transferred to water/steam in steam generator for each observed power plant. As the fuel mass flow rate and an exact fuel lower heating value was not known for both observed power plants, it is calculated the amount of energy transferred to water/steam in the steam generator for each observed power plant. Therefore, for the power plant without reheating, the energy transferred to water is calculated by using operating points 1 and 2, Figure 1 (a), while for the power plant with reheating, the energy transferred to the water/steam is calculated by using operating points 1 and 2 as well as operating points 6 and 7, Figure 1 (b). For a power plant with reheating, Figure 1 (b), cumulative energy transferred from fuel in a steam generator is the sum of energies transferred to water and to steam in the reheater.

The steam real (polytropic) expansion process of both observed marine propulsion steam turbines is presented in Figure 2. From Figure 2 it is clear the influence of the steam reheating process – although the steam reheater uses additional fuel for steam temperature (and consequentially steam-specific enthalpy) increases, the reheater retains the steam expansion process in the area of superheated steam as long as possible. Also, a steam reheater allows higher steam quality at the end of expansion in LPC (higher steam quality denotes higher steam content and fewer water droplets in the steam at the end of expansion in LPC). As the water droplets have a very erosive effect on the turbine blades, lowering water droplet content will notably extend maintenance or replacement periods for turbine stages that operate with wet steam.

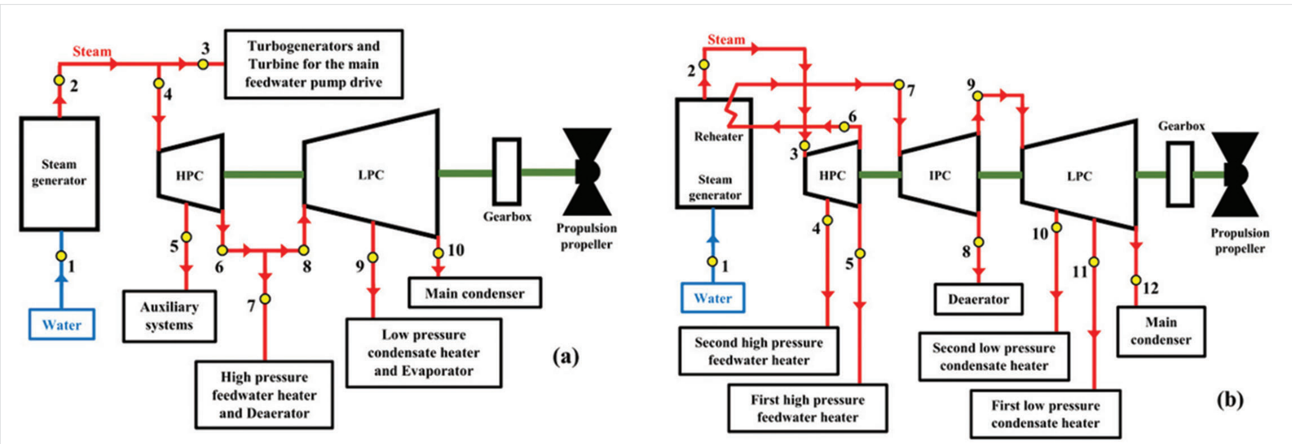

Figure 1 Simplified schemes of two observed marine propulsion steam turbines along with operating points required for the thermodynamic analysis:
(a) Marine steam turbine without reheating, (b) Marine steam turbine with reheating

Slika 1. Pojednostavnjena shema dviju promatranih brodskih propulzijskih parnih turbina s radnim točkama potrebnima za termodinamičku analizu:
(a) brodska parna turbina bez pregrijavanja pare, (b) brodska parna turbina s pregrijavanjem pare

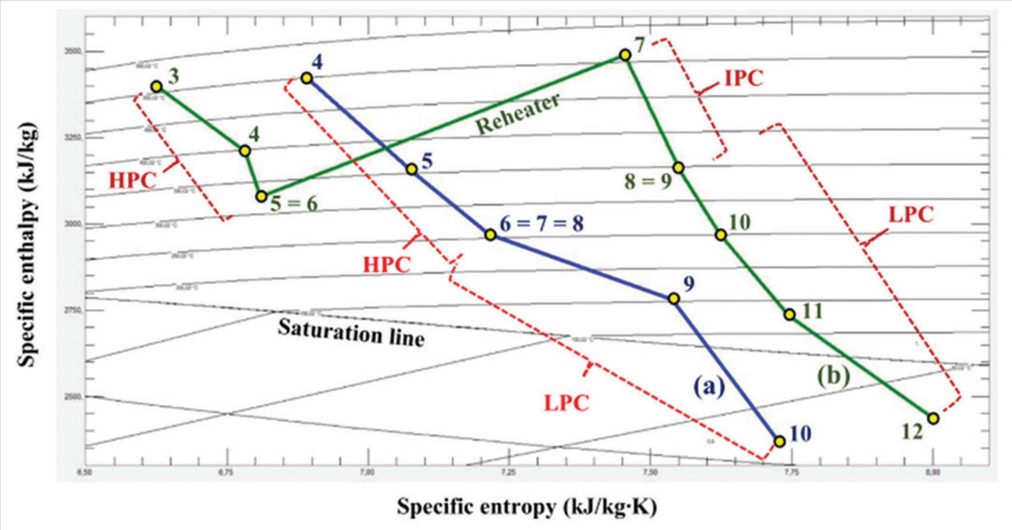

Figure 2 Steam real (polytropic) expansion process of both observed marine propulsion steam turbines: (a) Steam turbine without reheating - blue, (b) Steam turbine with reheating - green

Slika 2. Politropski stvarni proces ekspanzije obiju brodskih propulzijskih parnih turbina: (a) parna turbina bez pregrijavanja pare – plavo, b) parna turbina s pregrijavanjem pare – zeleno

3. ENERGY AND EXERGY ANALYSIS EQUATIONS / *Jednadžbe analize energije i eksergije*

3.1. Overall (general) equations and balances / *Ukupne (opće) jednadžbe i bilance*

In complete observation of any system or component, both energy and exergy analyses should be applied. The difference between these two analyses is that they observe different kind of losses which occurs during any system or component operation.

Energy analysis is based on the first law of thermodynamics and this analysis did not consider the conditions of the ambient inside which a system or a component operates [44]. In contrast to energy analysis, exergy analysis is based on the second law of thermodynamics and it considers the state of the ambient in which the observed system or a component operates [45]. Therefore, exergy analysis considers additional losses which are neglected in the energy analysis.

The literature review shows that the exergy analysis (due to the consideration of additional losses related to the ambient) is more used in the observation and optimization of many systems, components, or processes [46-48]. Also, an exergy analysis can be a baseline for further detail and more complex analyses [49-51].

In both energy and exergy analyses, there exists several overall (general) equations and balances which should always be satisfied, regardless of the observed system, process, or component. The first two such equations are general energy and exergy balances, which are defined according to [52, 53] as:

$$\dot{Q}_{\text{in}} + P_{\text{in}} + \sum \dot{En}_{\text{in}} - \dot{Q}_{\text{out}} - P_{\text{out}} - \sum \dot{En}_{\text{out}} = 0, \quad (1)$$

$$\dot{X}_{\text{in}} + P_{\text{in}} + \sum \dot{Ex}_{\text{in}} - \dot{X}_{\text{out}} - P_{\text{out}} - \sum \dot{Ex}_{\text{out}} = \dot{Ex}_{\text{L}}, \quad (2)$$

where $\dot{Q}$ is energy transfer by heat, P is mechanical power, subscript in denotes input (inlet), subscript out denotes output (outlet), and subscript L denotes loss.

$\dot{En}$ is the total energy flow of any fluid stream and $\dot{Ex}$ is the total exergy flow of any fluid stream. Both of these variables are defined according to [54, 55]:

$$\dot{En} = \dot{m} \cdot h, \quad (3)$$

$$\dot{Ex} = \dot{m} \cdot \varepsilon, \quad (4)$$

where $\dot{m}$ is fluid mass flow rate, h is fluid-specific enthalpy, while ε is fluid-specific exergy which definition can be found in [56] and presented by an equation:

$$\varepsilon = (h - h_0) - T_0 \cdot (s - s_0), \quad (5)$$

where s is fluid-specific entropy, T is fluid temperature and subscript 0 is related to the ambient state. The last undefined variable from the general exergy balance equation (Eq. 2) is an exergy transfer by heat at the temperature T ($\dot{X}$), which can be calculated according to [57, 58] by an equation:

$$\dot{X} = \sum(1 - \tfrac{T_0}{T}) \cdot \dot{Q}. \quad (6)$$

During any system or a component standard operation, fluid leakage did not occur. If there is no fluid leakage, always valid mass flow rate balance is [59]:

$$\sum \dot{m}_{\text{in}} = \sum \dot{m}_{\text{out}}. \quad (7)$$

The general energy or exergy efficiency equation, according to the literature [60, 61], is:

$$\eta_{\text{en (ex)}} = \frac{\text{cumulative energy (exergy) output}}{\text{cumulative energy (exergy) input}}. \quad (8)$$

However, it should be highlighted that the exact energy or exergy efficiency equation can be much different from the above presented general definition, which depends on the system or component characteristics and operation specificity.

3.2. Equations for the energy and exergy analyses of the observed marine propulsion steam turbines / *Jednadžbe za analizu energije i eksergije promatranih brodskih propulzijskih parnih turbina*

The equations used in the energy and exergy analyses of both observed marine propulsion steam turbines and their cylinders are presented in this subsection. These equations are developed according to recommendations from the literature [62-65]. It should be highlighted that all general equations and balances (presented in a previous subsection 3.1) are always satisfied, regardless of the observed turbine or turbine cylinder. Along with equations required for the analysis of both turbines and their cylinders, in this subsection, there are also presented equations for the calculation of whole power plant energy and exergy efficiency to obtain the influence of each observed steam turbine operation on the entire power plant.

For the exergy analysis of each observed marine steam turbine or any turbine cylinder, it is sufficient to know the operating fluid properties in a real (polytropic) expansion process – for each observed turbine that expansion process is presented in Figure 2.

However, for the energy analysis of any observed steam turbine or any turbine cylinder, the real (polytropic) expansion process is not sufficient. Energy analysis of any steam turbine or turbine cylinder is based on the comparison of real (polytropic) and ideal (isentropic) steam expansion processes. In comparison to the real (polytropic) steam expansion process, the ideal (isentropic) steam expansion process is the process between the same pressures, it uses the same mass flow rates, but in an ideal process fluid, specific entropy is always constant. Therefore, in an ideal steam expansion process are neglected all the losses which occur during real steam expansion are. Also, due to neglecting all expansion losses, the ideal (isentropic) steam expansion process in any turbine or cylinder will always result in higher-developed mechanical power in comparison to the real (polytropic) process. The ideal (isentropic) steam expansion process represents the maximal potential that can theoretically be obtained in any steam turbine or turbine cylinder. A comparison of ideal (isentropic) and real (polytropic) steam expansion processes in HPC of both observed marine propulsion steam turbines (with and without reheating) is presented in Figure 3 (operating points in Figure 3 are defined in accordance to Figure 1 and Figure 2). For any other cylinder of both observed turbines, the same logic and principle are valid. The operating points of the ideal (isentropic) process will be marked with a number (following Figure 1 and Figure 2) and with an addition of the word "is" – as presented in Figure 3.

Equations for the calculation of real (polytropic) developed mechanical power and ideal (isentropic) mechanical power of the whole turbine (WT) and each turbine cylinder are presented in Table 1 for the turbine without reheating and in Table 2 for the turbine with reheating. In all Tables from this subsection, index W denotes the turbine with reheating, and index WO denotes the turbine without reheating.

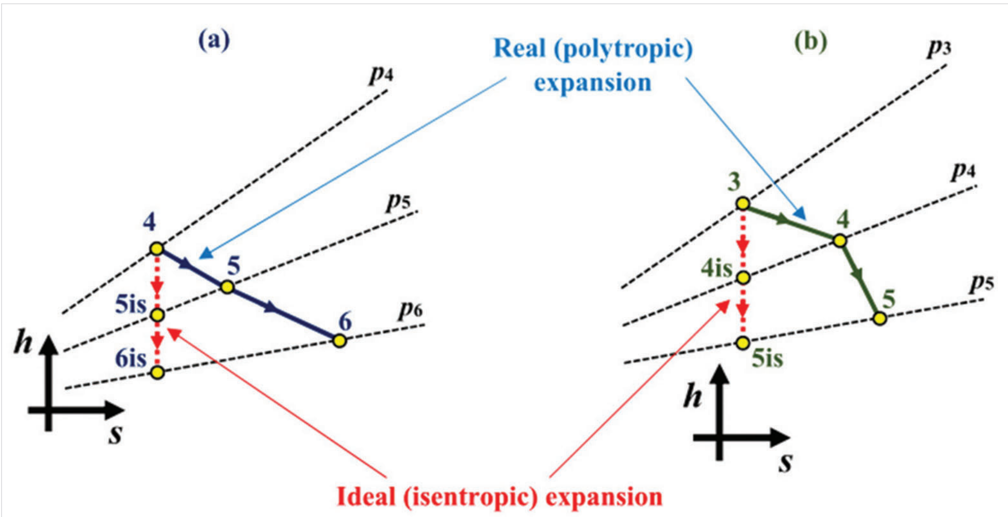

Figure 3 Comparison of ideal (isentropic) and real (polytropic) steam expansion processes in high-pressure cylinders (h-s diagram) of both observed turbines: (a) HPC of the marine steam turbine without reheating, (b) HPC of a marine steam turbine with reheating

Slika 3. Usporedba idealnoga (izenotropskog) i stvarnog (politropskog) procesa ekspanzije pare pri visokotlačnim kućištima (h-s dijagram) obiju promatranih turbina: (a) HPC brodske parne turbine bez pregrijavanja pare, (b) HPC brodske parne turbine s pregrijavanjem pare

Table 1 Equations for real (polytropic) and ideal (isentropic) mechanical power calculation of each cylinder and whole turbine without reheating

Tablica 1. Jednadžbe za stvarnu (politropsku) i idealnu (izenotropsku) kalkulaciju mehaničke snage svakoga kućišta i cijele turbine bez pregrijavanja pare

Component*	Mechanical power (real)	Eq.	Mechanical power (ideal)	Eq.
HPC	$P_{\mathrm{HPC,re,WO}} = \dot{m}_4 \cdot (h_4 - h_5) + (\dot{m}_4 - \dot{m}_5) \cdot (h_5 - h_6)$	(9)	$P_{\mathrm{HPC,id,WO}} = \dot{m}_4 \cdot (h_4 - h_{5\mathrm{is}}) + (\dot{m}_4 - \dot{m}_5) \cdot (h_{5\mathrm{is}} - h_{6\mathrm{is}})$	(12)
LPC	$P_{\mathrm{LPC,re,WO}} = \dot{m}_8 \cdot (h_8 - h_9) + (\dot{m}_8 - \dot{m}_9) \cdot (h_9 - h_{10})$	(10)	$P_{\mathrm{LPC,id,WO}} = \dot{m}_8 \cdot (h_8 - h_{9\mathrm{is}}) + (\dot{m}_8 - \dot{m}_9) \cdot (h_{9\mathrm{is}} - h_{10\mathrm{is}})$	(13)
WT	$P_{\mathrm{WT,re,WO}} = P_{\mathrm{HPC,re,WO}} + P_{\mathrm{LPC,re,WO}}$	(11)	$P_{\mathrm{WT,id,WO}} = P_{\mathrm{HPC,id,WO}} + P_{\mathrm{LPC,id,WO}}$	(14)

* Operating point enumeration is performed according to markings from Figure 1, Figure 2, and Figure 3.

Table 2 Equations for real (polytropic) and ideal (isentropic) mechanical power calculation of each cylinder and whole turbine with reheating

Tablica 2. Jednadžbe za stvarnu (politropsku) i idealnu (izentropsku) kalkulaciju mehaničke snage svakoga kućišta i cijele turbine s pregrijavanjem pare

Component*	Mechanical power (real)	Eq.	Mechanical power (ideal)	Eq.
HPC	$P_{\mathrm{HPC,re,W}} = \dot{m}_3 \cdot (h_3 - h_4) + (\dot{m}_3 - \dot{m}_4) \cdot (h_4 - h_5)$	(15)	$P_{\mathrm{HPC,id,W}} = \dot{m}_3 \cdot (h_3 - h_{4\mathrm{is}}) + (\dot{m}_3 - \dot{m}_4) \cdot (h_{4\mathrm{is}} - h_{5\mathrm{is}})$	(19)
IPC	$P_{\mathrm{IPC,re,W}} = \dot{m}_7 \cdot (h_7 - h_8)$	(16)	$P_{\mathrm{IPC,id,W}} = \dot{m}_7 \cdot (h_7 - h_{8\mathrm{is}})$	(20)
LPC	$P_{\mathrm{LPC,re,W}} = \dot{m}_9 \cdot (h_9 - h_{10}) + (\dot{m}_9 - \dot{m}_{10}) \cdot (h_{10} - h_{11}) + (\dot{m}_9 - \dot{m}_{10} - \dot{m}_{11}) \cdot (h_{11} - h_{12})$	(17)	$P_{\mathrm{LPC,id,W}} = \dot{m}_9 \cdot (h_9 - h_{10\mathrm{is}}) + (\dot{m}_9 - \dot{m}_{10}) \cdot (h_{10\mathrm{is}} - h_{11\mathrm{is}}) + (\dot{m}_9 - \dot{m}_{10} - \dot{m}_{11}) \cdot (h_{11\mathrm{is}} - h_{12\mathrm{is}})$	(21)
WT	$P_{\mathrm{WT,re,W}} = P_{\mathrm{HPC,re,W}} + P_{\mathrm{IPC,re,W}} + P_{\mathrm{LPC,re,W}}$	(18)	$P_{\mathrm{WT,id,W}} = P_{\mathrm{HPC,id,W}} + P_{\mathrm{IPC,id,W}} + P_{\mathrm{LPC,id,W}}$	(22)

* Operating point enumeration is performed according to markings from Figure 1, Figure 2, and Figure 3.

Energy loss and relative energy loss of each cylinder and whole turbine are calculated in the same manner for both observed turbines (with and without reheating) by using the equations presented in Table 3. For a steam turbine without reheating, which did not possess IPC, energy loss and relative energy loss of that cylinder are equal to zero.

Equations for exergy loss and relative exergy loss calculation of each cylinder and whole turbine are presented in Table 4 for a turbine without reheating and in Table 5 for a turbine with reheating.

Table 3 Equations for energy loss and relative energy loss calculation of each cylinder and whole turbine for both turbines (with and without reheating)

Tablica 3. Jednadžbe gubitka energije te izračun relativnoga gubitka energije svakoga kućišta i cijele turbine za obje turbine (s pregrijavanjem pare i bez njega)

Component	Energy loss	Eq.	Relative energy loss	Eq.
HPC	$\dot{E}n_{\mathrm{L,HPC}} = P_{\mathrm{HPC,id}} - P_{\mathrm{HPC,re}}$	(23)	$\dot{E}n_{\mathrm{L,HPC,Relative}} = \dfrac{\dot{E}n_{\mathrm{L,HPC}}}{P_{\mathrm{HPC,re}}} \cdot 100$	(27)
IPC	$\dot{E}n_{\mathrm{L,IPC}} = P_{\mathrm{IPC,id}} - P_{\mathrm{IPC,re}}$	(24)	$\dot{E}n_{\mathrm{L,IPC,Relative}} = \dfrac{\dot{E}n_{\mathrm{L,IPC}}}{P_{\mathrm{IPC,re}}} \cdot 100$	(28)
LPC	$\dot{E}n_{\mathrm{L,LPC}} = P_{\mathrm{LPC,id}} - P_{\mathrm{LPC,re}}$	(25)	$\dot{E}n_{\mathrm{L,LPC,Relative}} = \dfrac{\dot{E}n_{\mathrm{L,LPC}}}{P_{\mathrm{LPC,re}}} \cdot 100$	(29)
WT	$\dot{E}n_{\mathrm{L,WT}} = \dot{E}n_{\mathrm{L,HPC}} + \dot{E}n_{\mathrm{L,IPC}} + \dot{E}n_{\mathrm{L,LPC}}$	(26)	$\dot{E}n_{\mathrm{L,WT,Relative}} = \dfrac{\dot{E}n_{\mathrm{L,WT}}}{P_{\mathrm{WT,re}}} \cdot 100$	(30)

Table 4 Equations for exergy loss and relative exergy loss calculation of each cylinder and whole turbine without reheating

Tablica 4. Jednadžbe gubitka eksergije i izračun relativnoga gubitka eksergije za svako kućište i cijelu turbinu bez pregrijavanja pare

Component*	Exergy loss	Eq.	Relative exergy loss	Eq.
HPC	$\dot{E}x_{\mathrm{L,HPC,WO}} = \dot{E}x_4 - \dot{E}x_5 - \dot{E}x_6 - P_{\mathrm{HPC,re,WO}}$	(31)	$\dot{E}x_{\mathrm{L,HPC,Relative,WO}} = \dfrac{\dot{E}x_{\mathrm{L,HPC,WO}}}{P_{\mathrm{HPC,re,WO}}} \cdot 100$	(34)
LPC	$\dot{E}x_{\mathrm{L,LPC,WO}} = \dot{E}x_8 - \dot{E}x_9 - \dot{E}x_{10} - P_{\mathrm{LPC,re,WO}}$	(32)	$\dot{E}x_{\mathrm{L,LPC,Relative,WO}} = \dfrac{\dot{E}x_{\mathrm{L,LPC,WO}}}{P_{\mathrm{LPC,re,WO}}} \cdot 100$	(35)
WT	$\dot{E}x_{\mathrm{L,WT,WO}} = \dot{E}x_{\mathrm{L,HPC,WO}} + \dot{E}x_{\mathrm{L,LPC,WO}}$	(33)	$\dot{E}x_{\mathrm{L,WT,Relative,WO}} = \dfrac{\dot{E}x_{\mathrm{L,WT,WO}}}{P_{\mathrm{WT,re,WO}}} \cdot 100$	(36)

* Operating point enumeration is performed according to markings from Figure 1 and Figure 2.

Table 5 Equations for exergy loss and relative exergy loss calculation of each cylinder and whole turbine with reheating
Tablica 5. Jednadžbe eksergijskoga gubitka i izračun relativnoga gubitka eksergije svakoga kućišta i cijele turbine s pregrijavanjem pare

Component*	Exergy loss	Eq.	Relative exergy loss	Eq.
HPC	$\dot{E}x_{\mathrm{L,HPC,W}} = \dot{E}x_3 - \dot{E}x_4 - \dot{E}x_5 - \dot{E}x_6 - P_{\mathrm{HPC,re,W}}$	(37)	$\dot{E}x_{\mathrm{L,HPC,Relative,W}} = \frac{\dot{E}x_{\mathrm{L,HPC,W}}}{P_{\mathrm{HPC,re,W}}} \cdot 100$	(41)
IPC	$\dot{E}x_{\mathrm{L,IPC,W}} = \dot{E}x_7 - \dot{E}x_8 - \dot{E}x_9 - P_{\mathrm{IPC,re,W}}$	(38)	$\dot{E}x_{\mathrm{L,IPC,Relative,W}} = \frac{\dot{E}x_{\mathrm{L,IPC,W}}}{P_{\mathrm{IPC,re,W}}} \cdot 100$	(42)
LPC	$\dot{E}x_{\mathrm{L,LPC,W}} = \dot{E}x_9 - \dot{E}x_{10} - \dot{E}x_{11} - \dot{E}x_{12} - P_{\mathrm{LPC,re,W}}$	(39)	$\dot{E}x_{\mathrm{L,LPC,Relative,W}} = \frac{\dot{E}x_{\mathrm{L,LPC,W}}}{P_{\mathrm{LPC,re,W}}} \cdot 100$	(43)
WT	$\dot{E}x_{\mathrm{L,WT,W}} = \dot{E}x_{\mathrm{L,HPC,W}} + \dot{E}x_{\mathrm{L,IPC,W}} + \dot{E}x_{\mathrm{L,LPC,W}}$	(40)	$\dot{E}x_{\mathrm{L,WT,Relative,W}} = \frac{\dot{E}x_{\mathrm{L,WT,W}}}{P_{\mathrm{WT,re,W}}} \cdot 100$	(44)

* Operating point enumeration is performed according to markings from Figure 1 and Figure 2.

The energy and exergy efficiency of each cylinder and whole turbine are calculated in the same manner for both observed turbines (with and without reheating) by using the equations presented in Table 6. Steam turbines without reheating did not possess IPC, so both efficiencies for that cylinder are calculated only for the turbine with reheating.

As the exergy analysis is based on the ambient state in which the observed system or a component operates, any exergy analysis should be defined as the base ambient state. In this analysis, the base ambient state is defined according to recommendations from the literature [66] with ambient pressure equal to 1 bar and ambient temperature equal to 25 °C.

At the end of this analysis, it is performed the ambient temperature variation to observe the sensitivity of each turbine and turbine cylinder to the change in ambient parameters (ambient pressure remains always constant and equal to 1 bar). Exergy losses and exergy efficiencies for each ambient state are calculated by using the same equations presented in Table 4, Table 5, and Table 6.

Finally, this research it is also investigated how the operation of each observed marine steam turbine influences energy and exergy efficiencies of the entire power plant. Entire power plant efficiencies can be calculated only if the heat transferred from fuel to water/steam in steam generators is known. According to Figure 1, cumulative heat transferred from fuel to water in the steam generator of the power plant without reheating can be calculated as:

$$\dot{Q}_{\mathrm{WO}} = \dot{m}_1 \cdot (h_2 - h_1), \tag{53}$$

while for a power plant with steam reheating, heat transferred from fuel to water and steam in a steam generator can be calculated as:

$$\dot{Q}_{\mathrm{W}} = \dot{m}_1 \cdot (h_2 - h_1) + \dot{m}_6 \cdot (h_7 - h_6). \tag{54}$$

Both energy and exergy efficiencies of the entire power plant are calculated according to the equations presented in Table 7.

Table 6 Equations for energy and exergy efficiency calculation of each cylinder and whole turbine for both observed steam turbines (with and without reheating)
Tablica 6. Jednadžbe izračuna energijske i eksergijske iskoristivosti svakoga kućišta i cijele turbine za obje promatrane parne turbine (s pregrijavanjem pare i bez njega)

Component	Energy efficiency	Eq.	Exergy efficiency	Eq.
HPC	$\eta_{\mathrm{en,HPC}} = \frac{P_{\mathrm{HPC,re}}}{P_{\mathrm{HPC,id}}} \cdot 100$	(45)	$\eta_{\mathrm{ex,HPC}} = \frac{P_{\mathrm{HPC,re}}}{\dot{E}x_{\mathrm{L,HPC}} + P_{\mathrm{HPC,re}}} \cdot 100$	(49)
IPC	$\eta_{\mathrm{en,IPC}} = \frac{P_{\mathrm{IPC,re}}}{P_{\mathrm{IPC,id}}} \cdot 100$	(46)	$\eta_{\mathrm{ex,IPC}} = \frac{P_{\mathrm{IPC,re}}}{\dot{E}x_{\mathrm{L,IPC}} + P_{\mathrm{IPC,re}}} \cdot 100$	(50)
LPC	$\eta_{\mathrm{en,LPC}} = \frac{P_{\mathrm{LPC,re}}}{P_{\mathrm{LPC,id}}} \cdot 100$	(47)	$\eta_{\mathrm{ex,LPC}} = \frac{P_{\mathrm{LPC,re}}}{\dot{E}x_{\mathrm{L,LPC}} + P_{\mathrm{LPC,re}}} \cdot 100$	(51)
WT	$\eta_{\mathrm{en,WT}} = \frac{P_{\mathrm{WT,re}}}{P_{\mathrm{WT,id}}} \cdot 100$	(48)	$\eta_{\mathrm{ex,WT}} = \frac{P_{\mathrm{WT,re}}}{\dot{E}x_{\mathrm{L,WT}} + P_{\mathrm{WT,re}}} \cdot 100$	(52)

Table 7 Equations for the entire plant energy and exergy efficiency calculation
Tablica 7. Jednadžbe za izračun energijske i eksergijske iskoristivosti cijeloga postrojenja

	Plant energy efficiency	Eq.	Plant exergy efficiency	Eq.
Plant without reheating	$\eta_{\mathrm{en,Plant,WO}} = \frac{P_{\mathrm{WT,re,WO}}}{\dot{Q}_{\mathrm{WO}}} \cdot 100$	(55)	$\eta_{\mathrm{ex,Plant,WO}} = \frac{P_{\mathrm{WT,re,WO}}}{\dot{Q}_{\mathrm{WO}} \cdot \mathrm{EXC}} \cdot 100$	(57)
Plant with reheating	$\eta_{\mathrm{en,Plant,W}} = \frac{P_{\mathrm{WT,re,W}}}{\dot{Q}_{\mathrm{W}}} \cdot 100$	(56)	$\eta_{\mathrm{ex,Plant,W}} = \frac{P_{\mathrm{WT,re,W}}}{\dot{Q}_{\mathrm{W}} \cdot \mathrm{EXC}} \cdot 100$	(58)

In Eq. 57 and Eq. 58, is an exergy coefficient dependable on the fuel type. As both steam generators in both observed power plants use natural gas, in [67] can be found that the natural gas exergy coefficient is equal to 1.04 (based on the lower heating value).

It should also be highlighted that the equations presented in Table 7 did not consider the losses between fuel chemical energy and heat transferred to water/steam. The precise calculation will request that in the denominator of each equation from Table 7 instead of transferred heat should be fuel mass flow rate multiplied by a fuel lower heating value. Due to insufficient data, fuel mass flow rate and exact fuel lower heating value were not known for both steam generators, so the plant efficiencies are calculated by using transferred heat and neglecting heat losses during heat transfer.

4. FLUID PROPERTIES REQUIRED FOR THE ANALYSIS OF BOTH OBSERVED MARINE PROPULSION TURBINES / *Svojstva fluida potrebna za analizu obiju promatranih brodskih parnih turbina*

Fluid properties required for the energy and exergy analyses of each observed marine propulsion steam turbine are found in [31] for a turbine without reheating and presented in Table 8 and in [32] for a turbine with reheating and presented in Table 9. The fluid properties of each observed turbine are presented at nominal load. It should be highlighted that in the literature there are not found all fluid properties are presented in Table 8 and Table 9, in the literature there are found temperatures, pressures, and mass flow rates are only in each operating point of each observed turbine (Figure 1). Other fluid properties are calculated by using NIST-REFPROP 9.0 software [68].

Both Table 8 and Table 9 there are presented fluid properties of the real (polytropic) processes. It can be seen that the steam turbine with reheating has higher steam quality at the end of the expansion (0.95 in comparison to 0.92 for a steam turbine without reheating). Steam quality represents the steam percentage in the existing operating point at the end of the expansion (under the saturation line). Steam quality of 0.95 means that in such an operating point exists 95% of steam and 5% of water droplets. Steam quality equal to 1 denotes saturated steam which did not consist of any water droplet, while the steam quality of 0 denotes pure water.

Table 8 Steam properties in each operating point of the marine propulsion steam turbine without reheating
Tablica 8. Svojstva pare u svakoj radnoj točki brodske propulzijske parne turbine bez pregrijavanja

O. P.*	Temperature (°C)	Pressure (bar)	Mass flow rate (kg/s)	Specific enthalpy (kJ/kg)	Specific entropy (kJ/kg·K)	Quality	Specific exergy (kJ/kg)**
1	140	73.80	30.741	593.7	1.732	Subcooled	81.91
2	501	59.90	30.741	3425.6	6.887	Superheated	1376.90
3	500	59.00	3.942	3424.3	6.892	Superheated	1374.10
4	500	59.00	26.798	3424.3	6.892	Superheated	1374.10
5	350	15.65	0.908	3146.7	7.082	Superheated	1039.70
6	256	5.93	25.891	2970.4	7.213	Superheated	824.41
7	256	5.93	3.780	2970.4	7.213	Superheated	824.41
8	256	5.93	22.110	2970.4	7.213	Superheated	824.41
9	153	1.21	0.932	2781.1	7.538	Superheated	538.14
10	34.91	0.056	21.178	2370.9	7.726	0.92	72.12

* O. P. = Operating Point (following Figure 1)
** Presented specific exergies in each operating point are calculated for the base ambient state

Table 9 Steam properties in each operating point of the marine propulsion steam turbine with reheating
Tablica 9. Svojstva pare brodske propulzijske parne turbine s pregrijavanjem u svakoj radnoj točki

O. P.*	Temperature (°C)	Pressure (bar)	Mass flow rate (kg/s)	Specific enthalpy (kJ/kg)	Specific entropy (kJ/kg·K)	Quality	Specific exergy (kJ/kg)**
1	241.72	110.00	15.593	1046.6	2.701	Subcooled	245.75
2	512.85	103.00	15.593	3404.6	6.625	Superheated	1434.00
3	509.85	101.00	15.593	3399.3	6.626	Superheated	1428.20
4	397.86	38.70	1.055	3211.7	6.782	Superheated	1194.30
5	326.80	22.60	1.679	3079.2	6.809	Superheated	1053.80
6	326.80	22.60	12.859	3079.2	6.809	Superheated	1053.80
7	509.85	20.30	12.859	3489.7	7.455	Superheated	1271.70
8	341.80	5.60	0.421	3149.8	7.553	Superheated	902.38
9	341.80	5.60	12.438	3149.8	7.553	Superheated	902.38
10	249.77	2.40	0.808	2969.4	7.623	Superheated	701.18
11	126.82	0.60	0.683	2734.1	7.746	Superheated	429.35
12	32.87	0.050	10.947	2439.6	7.998	0.95	59.56

* O. P. = Operating Point (following Figure 1)
** Presented specific exergies in each operating point are calculated for the base ambient state

5. RESULTS AND DISCUSSION / *Rezultati i rasprava*

The real developed mechanical power of each cylinder and whole turbine for both observed marine propulsion steam turbines is presented in Figure 4.

From Figure 4 it can be seen that mechanical power developed by whole turbines or each cylinder is not directly comparable. This is why all the losses (both energy and exergy) will be presented in relative form – by the unit of the produced mechanical power.

Each cylinder of a steam turbine with reheating produces much lower mechanical power in comparison to any cylinder of a steam turbine without reheating. Consequentially produced mechanical power of the whole turbine is lower for a turbine with reheating (equal to 17426.55 kW) than for a turbine without reheating (24876.55 kW).

By observing real developed mechanical power in turbine cylinders, it can be concluded that both cylinders of the steam turbine without reheating (HPC and LPC) at nominal load develop very similar mechanical power, while for a steam turbine with reheating developed mechanical power notably varies from one cylinder to another. At nominal load, the turbine with reheating develops the lowest mechanical power in IPC, followed by HPC, while its LPC develops mechanical power only slightly lower than both HPC and IPC cumulatively.

Finally, observing all cylinders of both turbines, it can be concluded that the highest mechanical power in both turbines is produced in the last cylinder (LPC), although at least the last few LPC stages operate by using wet steam which increases cylinder losses (all other cylinders operate by using superheated steam).

Figure 5 presents the relative energy loss of each cylinder and whole turbine for both observed marine propulsion steam turbines (with and without reheating).

HPC of both observed turbines has the highest relative energy loss, much higher in comparison to the other cylinders. The highest energy loss in HPC of both turbines can be explained by the highest pressure and temperature of steam which expands through that cylinder (much higher in comparison to other cylinders). HPC of both turbines also has the characteristic that relative energy loss is only slightly higher for a steam turbine without reheating in comparison to a turbine with reheating (34.92% in comparison to 33.72%).

The relative energy loss of LPC for both turbines is notably lower when compared to HPC. Also, for LPC it can be seen that the relative energy loss of a turbine without reheating is notably higher in comparison to the turbine with reheating. The lowest relative energy loss of all cylinders has the IPC of a turbine with reheating (equal to 17.45%). From the relative energy loss viewpoint only, it can be concluded that IPC operation is nearest to the optimal.

Observing whole marine steam turbines, it is clear that the turbine without reheating has notably higher relative energy loss (equal to 30.77%) in comparison to the turbine with reheating whose relative energy loss is 22.77%.

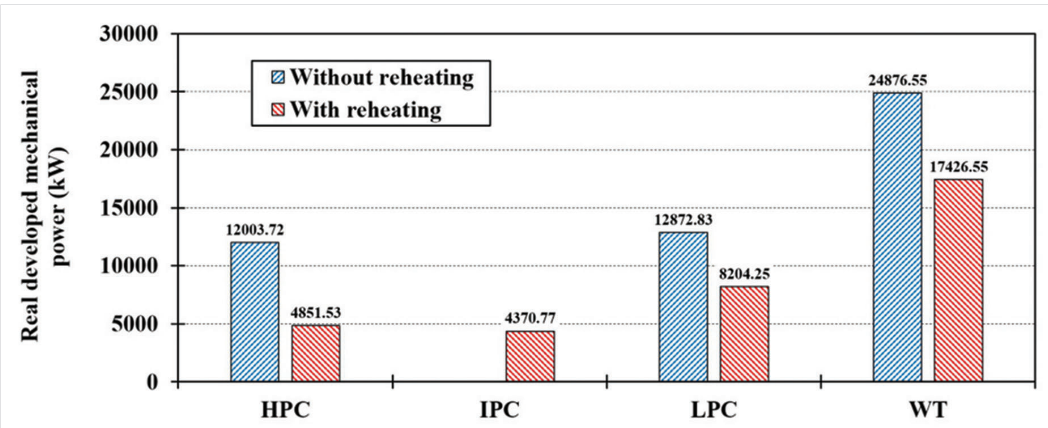

Figure 4 Real developed mechanical power of each cylinder and whole turbine for both observed marine propulsion steam turbines (with and without reheating)

Slika 4. Stvarno razvijena mehanička snaga svakoga kućišta i cijele turbine za obje promatrane brodske parne turbine (s pregrijavanjem pare i bez njega)

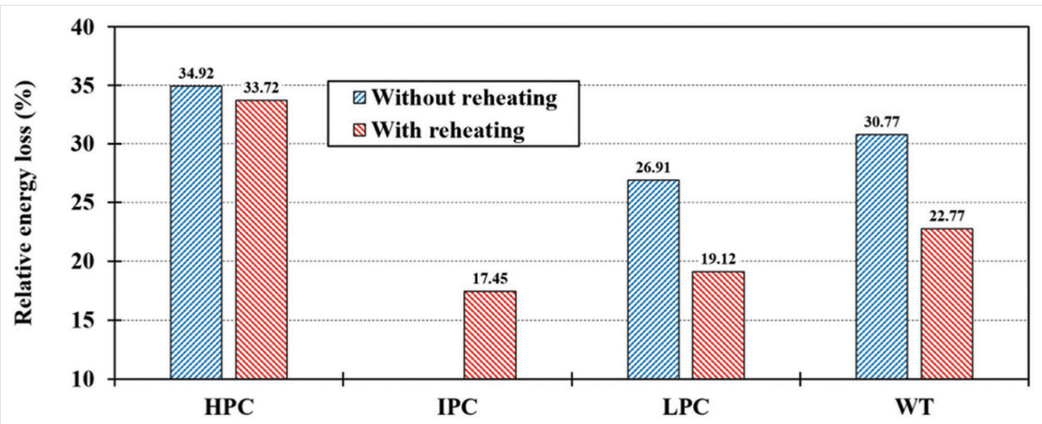

Figure 5 Relative energy loss of each cylinder and whole turbine for both observed marine propulsion steam turbines (with and without reheating)

Slika 5. Relativan gubitak energije svakoga kućišta i cijele turbine za obje promatrane parne turbine (s pregrijavanjem pare i bez njega)

The energy efficiency of each cylinder and whole turbine for both observed marine propulsion steam turbines is presented in Figure 6.

Due to high steam temperatures and pressures HPC of both observed steam turbines has much lower energy efficiency in comparison to all other cylinders. For the LPC of both steam turbines is easily noticeable that the energy efficiency of the steam turbine with reheating is much higher than the energy efficiency of the steam turbine without reheating.

From the energy viewpoint, the IPC of the turbine with reheating is the best-balanced cylinder of all cylinders (it has the lowest relative energy loss and the highest energy efficiency). Such a result can be explained by the fact that IPC did not operate with the steam of the highest temperature and pressure (as HPC) and simultaneously IPC did not operate with wet steam (as LPC) but completely by using superheated steam. Steam of the highest temperature and pressure, as well as wet steam, are the results of increased relative energy loss in HPC and LPC and thus lower energy efficiency in comparison to IPC. IPC energy efficiency is equal to 85.15%, which is very high energy efficiency for any marine steam turbine cylinder.

A whole turbine with reheating has much higher energy efficiency in comparison to a whole turbine without reheating (81.46% in comparison to 76.47%), which is a confirmation that the steam reheating process is very beneficial for the turbine (and its cylinders) operation.

A comparison of Figure 5 and Figure 6 shows that for all the cylinders and the whole turbine, regardless of which turbine is considered, relative energy loss and energy efficiency are reverse proportional – higher relative energy loss will result in lower energy efficiency and vice versa.

It should also be highlighted that marine steam turbines (main propulsion turbines or auxiliary ones) and their cylinders have much lower efficiencies (both energy and exergy) in comparison to steam turbines from various conventional steam power plants [69]. There are several reasons for such an occurrence. First of all, marine steam turbines develop much lower mechanical power in comparison to steam turbines from conventional power plants. As the steam turbine efficiency decreases with the decrease in developed mechanical power (due to increased losses per unit of produced power), lower efficiencies of marine steam turbines can be expected. Secondly, all marine steam turbines and their cylinders (main propulsion turbines or auxiliary ones) must be able to accept various and frequent load changes (dynamic operation), as requested by current ship procedures and processes. Dynamic loads will also decrease the efficiencies of turbines and their cylinders [70-72].

Exergy analysis, as mentioned before, considers a different kind of loss in comparison to energy analysis. However, almost all main conclusions related to relative exergy loss, Figure 7, remain the same as for relative energy loss, Figure 5.

The only noticeable difference between relative energy and exergy losses related to the cylinders of the observed marine propulsion steam turbines is that from the exergy viewpoint, the cylinder with the highest relative exergy loss is LPC, regardless of which of the two analyzed steam turbines is observed. Therefore, it can be concluded that in the exergy analysis, wet steam losses are more influential than losses related to steam of high temperature and pressure used in HPC. Figure 7 also clear that both HPC and LPC of the steam turbine without reheating have notably higher relative exergy loss in comparison to the same cylinders from the turbine with reheating.

IPC of the turbine with reheating did not have the lowest relative energy loss only, it also has the lowest relative exergy loss, much lower in comparison to all other cylinders, Figure 5 and Figure 7. A whole marine propulsion steam turbine without reheating has notably higher relative exergy loss (equal to 23.55%) in comparison to a whole propulsion steam turbine with reheating (whole steam turbine with reheating has relative exergy loss equal to 15.63%).

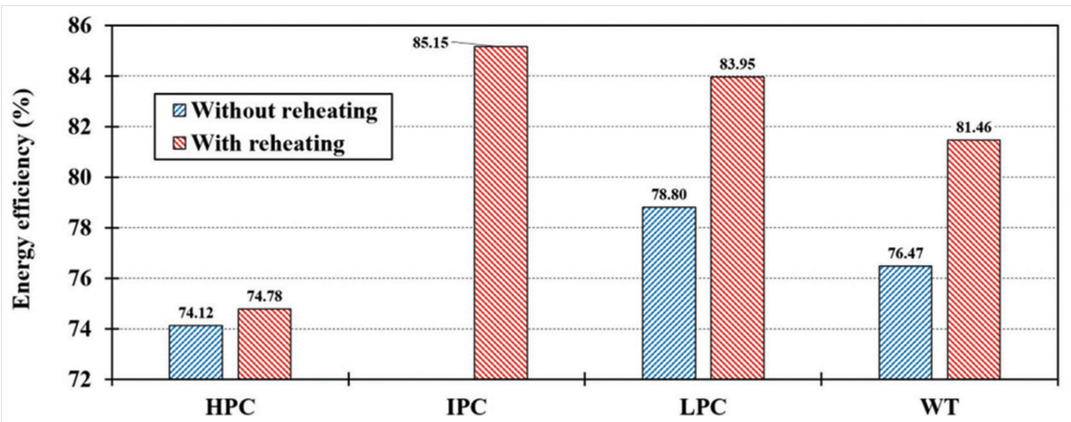

Figure 6 Energy efficiency of each cylinder and whole turbine for both observed marine propulsion steam turbines (with and without reheating)

Slika 6. Energetska iskoristivost svakoga kućišta i cijele turbine za obje promatrane brodske propulzijske parne turbine (s pregrijavanjem pare i bez njega)

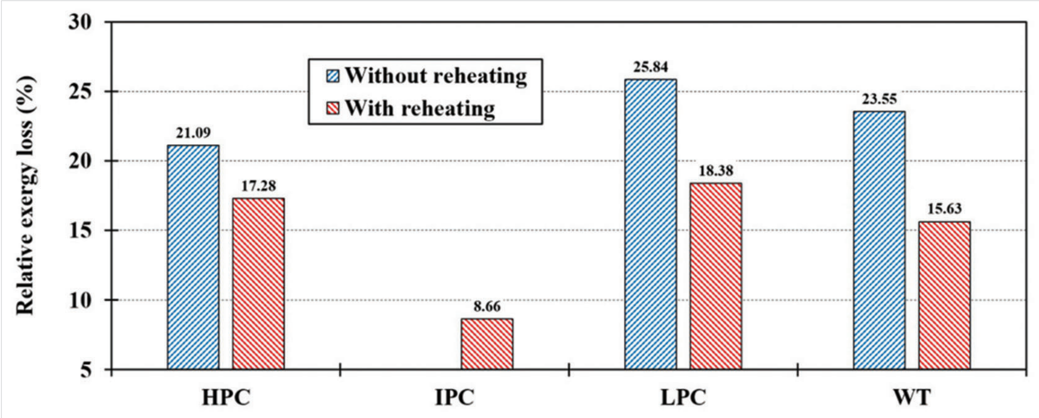

Figure 7 Relative exergy loss of each cylinder and whole turbine for both observed marine propulsion steam turbines (with and without reheating)

Slika 7. Relativan gubitak eksergije svakoga kućišta i cijele turbine za obje promatrane brodske propulzijske parne turbine (s pregrijavanjem pare i bez njega)

A comparison of Figure 7 and Figure 8 shows that relations valid between relative energy loss and energy efficiency are identical for exergy analysis parameters. For both observed marine steam turbines and their cylinders it is valid that relative exergy loss and exergy efficiency are reverse proportional – higher relative exergy loss will result in lower exergy efficiency and vice versa.

Observing the cylinders of both marine steam turbines, it can be concluded that LPC has the lowest exergy efficiency, followed by HPC. For the IPC of a steam turbine with reheating, both energy and exergy analyses show that this cylinder has the lowest relative energy and exergy losses as well as the highest efficiencies (both energy and exergy) of all cylinders. The exergy efficiency of IPC is equal to 92.03%, which is very high exergy efficiency, comparable to the cylinders of steam turbines from conventional power plants. Such high IPC efficiencies (both energy and exergy) prove that this cylinder operates in the best possible conditions in comparison to the other cylinders.

Due to the much higher relative exergy loss of the whole steam turbine without reheating, Figure 7, this turbine has consequentially much lower exergy efficiency in comparison to the turbine with reheating (80.94% in comparison to 86.48%), Figure 8. Also, the exergy analysis shows the benefits of the steam reheating process – a turbine with reheating has a lower relative exergy loss and higher exergy efficiency in comparison to a turbine without reheating, which is valid not just for the whole turbine, but also for the turbine cylinders.

This analysis also performed the variation of the ambient temperature to examine exergy parameters sensitivity to the ambient temperature change for both marine steam turbines and their cylinders. The ambient temperature is varied in the real expected range from 5 °C up to 45 °C (in steps of 10 °C), while the ambient pressure remains always the same and equal as at the base ambient state (1 bar).

Figure 9 it is presented the average change in relative exergy loss during the ambient temperature variation of each cylinder and whole turbine for both observed marine propulsion steam turbines. From Figure 9 it is clear that in terms of relative exergy loss, cylinders of a steam turbine without reheating (HPC and LPC) are much more sensitive to the ambient temperature change in comparison to the same cylinders from a turbine with reheating. Observing all the cylinders, it can be concluded that the LPC of both analyzed turbines is the cylinder that is the most sensitive to the ambient temperature change in terms of relative exergy loss. Relative exergy loss of the IPC from a turbine with reheating is the lowest influenced by the ambient temperature change of all cylinders (considering both analyzed steam turbines).

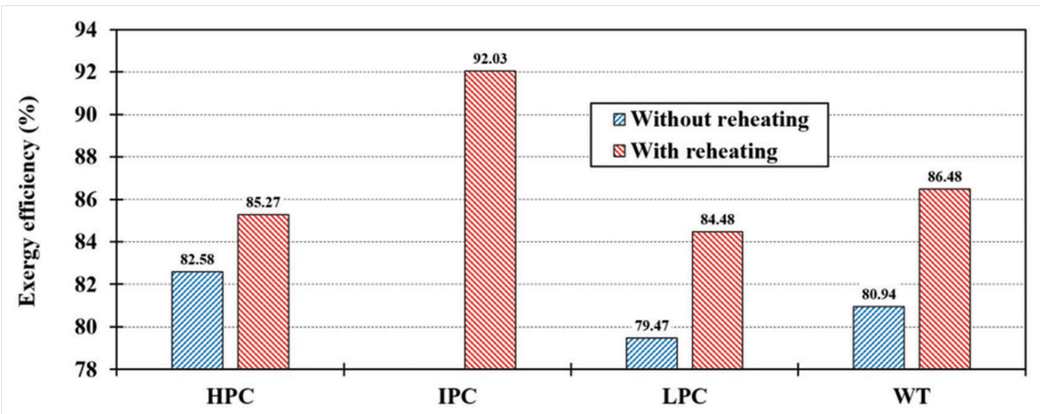

Figure 8 Exergy efficiency of each cylinder and whole turbine for both observed marine propulsion steam turbines (with and without reheating)

Slika 8. Eksergijska iskoristivost svakoga kućišta i cijele turbine za obje promatrane propulzijske parne turbine (s pregrijavanjem pare i bez njega)

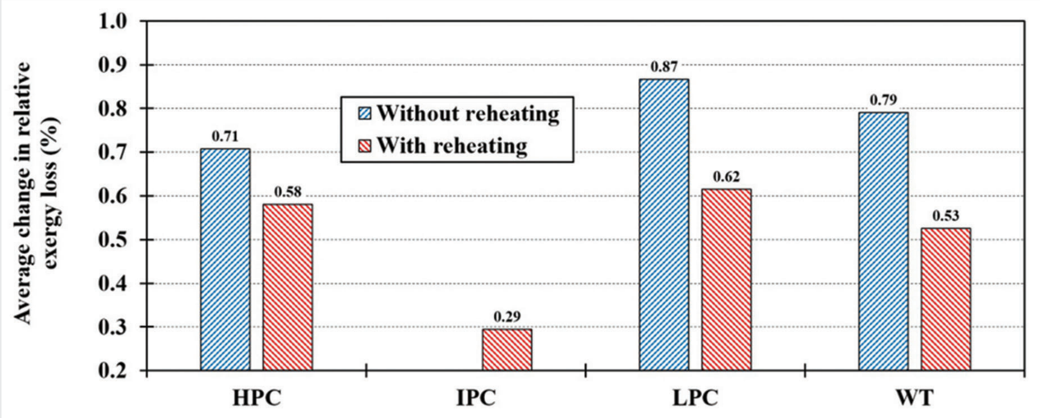

Figure 9 Average change in relative exergy loss during the ambient temperature variation of each cylinder and whole turbine for both observed marine propulsion steam turbines (with and without reheating)

Slika 9. Prosječna promjena u relativnom eksergijskom gubitku tijekom ambijentalnih promjena temperature u svakome kućištu i cijeloj turbini za obje promatrane brodske propulzijske parne turbine (s pregrijavanjem pare i bez njega)

Observing whole turbines, the relative exergy loss of steam turbine without reheating is much more influenced by the ambient temperature change in comparison to the whole steam turbine with reheating (the average change in relative exergy loss between ambient temperatures 5 °C and 45 °C is equal to 0.79% for a turbine without reheating and 0.53% for a turbine with reheating).

In terms of relative exergy loss, it can be concluded that the whole steam turbine without reheating as well as its cylinders are much more influenced by the ambient temperature change in comparison to the whole steam turbine with reheating and its cylinders.

The average change in exergy efficiency during the ambient temperature variation of each cylinder and whole turbine for both observed marine propulsion steam turbines is presented in Figure 10. The average change in exergy efficiency shows an identical trend as the average change in relative exergy loss during the ambient temperature variation, Figure 9 and Figure 10.

The Exergy efficiency of cylinders and the whole turbine without reheating is much more sensitive to the ambient temperature change in comparison to the whole steam turbine with reheating and its cylinders. For both observed steam turbines, LPC is a cylinder whose exergy efficiency is the most sensitive to the ambient temperature change, while the IPC of a steam turbine with reheating is the cylinder whose exergy efficiency is the lowest influenced by the ambient temperature change.

For whole observed steam turbines, the average change in exergy efficiency during the ambient temperature variation is notably higher for steam turbines without reheating in comparison to a steam turbine with reheating (0.52% in comparison to 0.39%).

Finally, it can be concluded that the exergy parameters (relative exergy loss and exergy efficiency) of a steam turbine without reheating as well as its cylinders are much more influenced by the ambient temperature change in comparison to the steam turbine with reheating and its cylinders.

At the end of this analysis, as the energy transferred from fuel to water/steam in a steam generator is known, it is calculated energy and exergy efficiency of the entire power plants in which observed turbines operate. Power plant energy and exergy efficiencies, for both observed steam turbines are presented in Figure 11. It should be highlighted that for both observed steam turbines and power plants (which did and did not possess steam reheater), the energy transferred to water/steam in the steam generator is lower than chemical energy contained in the fuel, therefore for both power plants calculated energy and exergy efficiencies will be slightly higher than the real ones. As this analysis intends to observe the influence of the

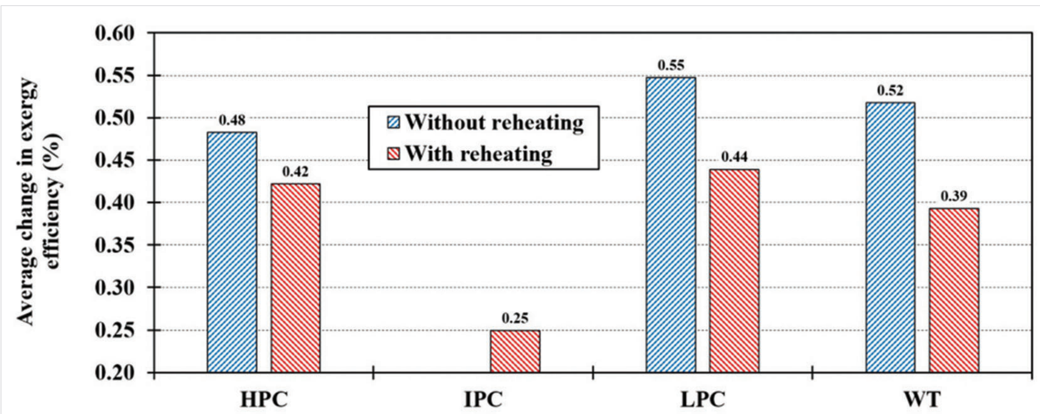

Figure 10 Average change in exergy efficiency during the ambient temperature variation of each cylinder and whole turbine for both observed marine propulsion steam turbines (with and without reheating)

Slika 10. Prosječna promjena eksergijske iskoristivosti tijekom varijacija ambijentalne temperature svakoga kućišta i cijele turbine za obje promatrane brodske propulzijske parne turbine (s pregrijavanjem pare i bez njega)

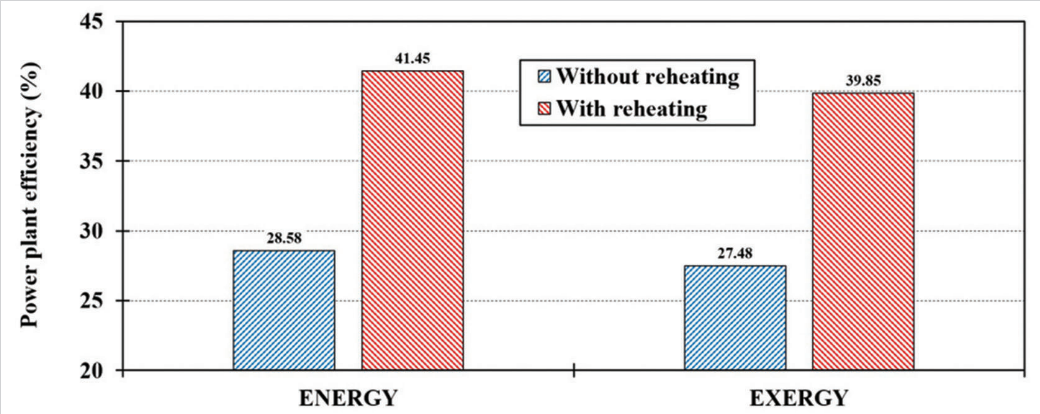

Figure 11 Marine steam power plant energy and exergy efficiency in two different arrangements: when using a turbine with reheating and when using a turbine without reheating

Slika 11. Energetska i eksergijska iskoristivost brodskog parnog postrojenja u dvama slučajevima: kada se koristi turbina s pregrijavanjem pare i kada se koristi turbina bez pregrijavanja pare

steam reheating process on the turbine, turbine cylinders, and entire power plant operation, heat losses in a steam generator can be neglected, while obtaining results of the power plant efficiency (both energy and exergy) are directly comparable.

From Figure 11 it is clear that the steam reheating process did not increase efficiencies and reduce losses of a steam turbine and its cylinders only, a steam reheating process also notably increases power plant efficiency (both energy and exergy) in comparison to the power plant which did not possess steam reheater. The steam reheating process will increase the efficiencies of the whole power plant (both energy and exergy) between 10% and 12% in real exploitation conditions, Figure 11.

Further research related to the analyzed marine propulsion steam turbines can be performed in several different ways. It will surely be interesting to investigate the improvement possibilities for each steam turbine and its cylinders. Also, each turbine can be performed various optimizations using conventional [73] or artificial intelligence methods and techniques, which show its potential not only in the marine sector [74, 75], but also in many other applications and processes [76, 77].

6. CONCLUSIONS / *Zaključci*

This research it is performed thermodynamic (energy and exergy) analysis and comparison of two marine propulsion steam turbines based on their operating parameters from exploitation. The first turbine doesn't possess steam reheating and has two cylinders (HPC and LPC), while the second turbine possesses steam reheating and has one additional cylinder (IPC). In the existing literature, there cannot be found at the moment a direct and exact comparison of two observed steam turbines and their cylinders or the exact benefits which the steam reheating process brings to the steam turbine and power plant operation. Moreover, all recommendations from the literature are only general so far, while in this research there are obtained and presented exact recommendations achievable in the real exploitation conditions.

Relative energy and exergy losses as well as energy and exergy efficiencies are calculated and compared for whole turbines and each cylinder of both observed turbines. It investigated the sensitivity of exergy parameters (relative exergy loss and exergy efficiency) to the ambient temperature change for both turbines and each cylinder. In the end, it is calculated and presented the influence of the steam reheating process on the energy and exergy efficiency of the entire power plant. The main conclusions of the performed analysis are:

- The highest mechanical power in both analyzed turbines is produced in the last, low-pressure cylinder, even though at least the last few stages of that cylinder operate by using wet steam which increases cylinder losses (all the other cylinders operate by using superheated steam).
- For both observed turbines and their cylinders it is valid that relative losses and efficiencies (both energy and exergy) are reverse proportional – an increase in relative loss results in a decrease in efficiency and vice versa.
- Both energy and exergy analyses show that IPC is a cylinder that avoids the dominant losses in the turbine and that its operation is the closest to optimal.
- Due to different origins of losses which are considered in energy and exergy analyses, in both observed turbines energy analysis detects HPC as the most problematic cylinder (due to its operation with a steam of the highest temperature and pressure), while exergy analysis detects LPC as the most problematic cylinder (due to its operation by using wet steam).
- The steam reheating process decreases losses and increases efficiencies (both energy and exergy) of each turbine cylinder and the whole turbine.
- The whole observed turbine with reheating has much higher efficiencies (both energy and exergy) in comparison to a turbine that did not possess steam reheating. The whole turbine with reheating has an energy efficiency equal to 81.46% and an exergy efficiency equal to 86.48%, while the whole turbine without reheating has energy and exergy efficiencies equal to 76.47% and 80.94%, respectively.
- The average change in exergy efficiency during the ambient temperature variation shows the identical trend as the average change in relative exergy loss for both observed turbines and their cylinders. Exergy parameters (relative exergy loss and exergy efficiency) of a steam turbine without reheating as well as its cylinders are much more influenced by the ambient temperature change in comparison to the steam turbine with reheating and its cylinders.
- The steam reheating process did not increase efficiencies and reduce losses of a steam turbine and its cylinders only,

a steam reheating process also notably increases power plant efficiency in comparison to the power plant which did not possess a steam reheater. The steam reheating process will increase efficiencies of the whole power plant (both energy and exergy) between 10% and 12% in real exploitation conditions.

Author Contributions: Conceptualization, I. P. and V. M.; Methodology, I. P.; Software, V. M.; Validation, I. P. and V. M.; Formal Analysis, V. M.; Investigation, I. P.; Resources, I. P. and V. M.; Data Curation, I. P.; Writing – Original Draft Preparation, V. M.; Writing – Review & Editing, I. P.; Visualization, V. M.; Supervision, I. P.; Project Administration, I. P.

Funding: The research presented in the manuscript did not receive any external funding.

Conflict of interest: None.

Acknowledgments: This research has been supported by the Croatian Science Foundation under the project IP-2018-01-3739, the University of Rijeka scientific grant uniri-tehnic-18-18-1146, and the University of Rijeka scientific grant uniri-tehnic-18-14.

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