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Thermodynamic analysis of an innovative steam turbine power plant with oxy-methane combustors

VLADIMIR OLEGOVICH KINDRA*
SERGEY KONSTANTINOVICH OSIPOV
OLGA VLADIMIROVNA ZLYVKO
IGOR ALEXANDROVICH SHCHERBATOV
VLADIMIR PETROVICH SOKOLOV

National Research University "Moscow Power Engineering Institute", Krasnokazarmennaya 14, Moscow, 111250 Russia

Abstract The Rankine cycle steam turbine power plants make a base for world electricity production. The efficiency of modern steam turbine units is not higher than 43–45%, which is remarkably lower compared to the combined cycle power plants. However, an increase in steam turbine power plant efficiency could be achieved by the rise of initial cycle parameters up to ultra-supercritical values: 700–780°C, 30–35 MPa. A prospective steam superheating technology is the oxy-fuel combustion heating in a side-mounted combustor located in the steam pipelines. This paper reviews thermal schemes of steam turbine power plants with one or two side-mounted steam superheaters. An influence of the initial steam parameters on the facility thermal efficiency was identified and primary and secondary superheater parameters were optimized. It was found that the working fluid superheating in the side-mounted oxy-methane combustors leads to an increase of thermal efficiency higher than that with the traditional boiler superheating in the initial temperature ranges of 700–780°C and 660–780°C by 0.6% and 1.4%, respectively.

Keywords: Combined cycle power plant; Carbon capture and storage system; Pre-combustion capture; Post-combustion capture; Oxy-fuel combustion

*Corresponding Author. Email: kindra.vladimir@yandex.ru

Abbreviations

ASU	–	air separation unit
HPT, IPT, LPT	–	high-, intermediate-, low-pressure turbine
STU	–	steam turbine units
TPP	–	thermal power plant

1 Introduction

Sustainable development of the power industry requires a transition to highly efficient and environmentally safe technology of electricity production from organic fuels. Rankine cycle steam turbine units (STU) are the base of the world power industry. Widespread use of this type of unit is due to their capability of different fuels operation, a wide range of power production and long life. Nowadays the best steam turbine units have net efficiency below 43–45%. In turn, the advanced combined cycle power plants have thermal efficiency above 60% [1, 2]. The current trend is a subsequent increase of the STU installed power (Fig. 1). By 2040 it will be 30% higher [3]. Therefore, the improvement of the STU thermal efficiency is an essential goal. It will provide not only the preservation of the primary resource but will also mitigate such harmful atmospheric emissions as carbon dioxide and nitrogen oxides.

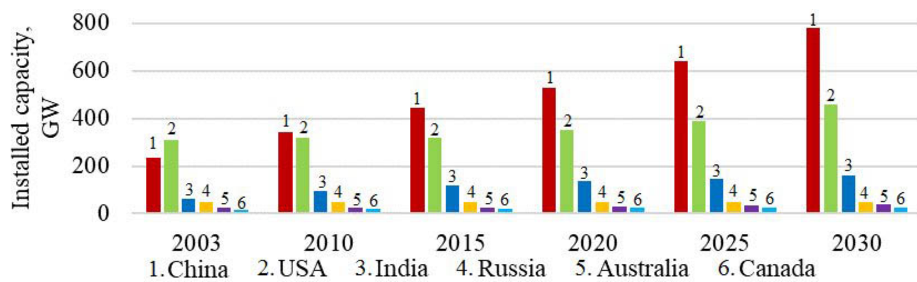


Figure 1: Growth dynamics of the thermal power plants installed capacity.

The remarkable STU efficiency increase may be reached through the cycle initial parameters increase and specifically by a transition from supercritical (540°C, 24 MPa) to the high supercritical (600–650°C, 28–30 MPa) and the ultra-supercritical (700–780°C, 32–35 MPa) steam parameters, which is followed by the thermal efficiency increase of 8–9% [4–6].

Numerous papers disclose the optimization of structure and thermodynamic parameters of steam turbine power plants with high steam parameters [7–10]. Particularly, paper [7] proposes a new flow chart for a 1000 MW STU with high supercritical steam parameters and single reheat with the following parameters: the initial and reheat temperatures of 600°C and 610°C, respectively, and the initial pressure of 301 bar. The feeding water regenerative heating system consists of four high-pressure heaters, a deaerator and five low-pressure heaters. Nine additional air heaters improved the cycle efficiency. In these heaters, the air is heated with the turbine bleeding steam before it enters the boiler. The exiting steam is supplied to the regenerative system heaters. This technical solution increases the STU net efficiency from 45.55 to 46.16%. Paper [8] describes a thermodynamic analysis and structure optimization for a 1000 MW STU with high supercritical parameters and double reheat with the following parameters: the initial and double reheat temperatures of 600°C and 610°C, respectively, and the initial pressure of 30 MPa. The feeding water regenerative heating system consists of three high-pressure heaters, a deaerator and five low-pressure heaters. Two additional external steam coolers improve the cycle net efficiency up to 46.99%. Paper [9] also describes a 1000 MW supercritical STU thermodynamic analysis. The STU has single reheat in its boiler. The initial and reheat steam temperatures were 600°C and 610°C, respectively, and the initial pressure was 27 MPa. The feeding water regeneration system has three high-pressure heaters, a deaerator and five low-pressure heaters. The heat of the boiler economizer exit gas is used for heating a part of condensate. The special turbine bleeding heats the air before the fuel combustion. These measures allowed one to achieve a net efficiency of 47.02%. However, this estimation does not include the pump compression losses. Paper [10] presents a techno-economic analysis of 900 MW STU with high supercritical and ultra-supercritical steam parameters. The STU with single reheat and high supercritical parameters has the initial and reheat steam temperatures of 650°C and 670°C, respectively, and initial pressure of 30 MPa. The STU with single reheat and ultra-supercritical parameters has the initial and reheat steam temperatures of 700°C and 720°C, respectively, and initial pressure of 35 MPa. The STU with double reheat and ultra-supercritical parameters has the first and secondary reheat temperatures of 720°C and initial pressure of 37.5 MPa. There were three high-pressure heaters, a deaerator and four low-pressure heaters in the feeding water regeneration system.

The ultra-supercritical power unit with single reheat has three high-pressure heaters, a deaerator and five low-pressure heaters, but the number of low-pressure heaters is increased up to 6 for the version with double reheat. The high supercritical power unit net efficiency is 45.52%, with ultra-supercritical parameters and single reheat 47.3% and with double reheat 48.13%. On the other side, the transition to higher steam parameters is followed by the 5–12% higher thermal power plant (TPP) construction capital investment.

The major influence upon the equipment price have construction expenses for the boiler, main steam pipelines and the turbine flowpath [11–14]. The flowpath cost increase is mostly due to the larger share of nickel alloys. In some cases, it might be possible to retain the low-pressure turbine by its modification [15].

The equipment prime cost may be reduced by the transition to the horizontal boiler layout or by steam overheating above 540°C in the side-mounted combustion chambers, or by fuel combustion in the working fluid flow [16, 17]. This allows retaining the boiler structure without changes, avoiding the use of nickel alloys for steam pipelines, and reducing the length of the primary and secondary steam pipelines that may be mostly manufactured of traditional materials.

The side-mounted combustion chambers may burn natural gas or hydrogen. The hydrogen employment requires manufacturing and storage facilities, which remarkably increase the TPP construction cost and the domestic power consumption. Some of the specific hydrogen fuel problems may be solved by the transition to oxy-methane fuel for the side-mounted combustion chambers. This solution reduces the fuel preparation and compression losses and does not require natural gas storage because methane is supplied to TPP by a pipeline. In this case, methane combustion produces non-condensing carbon dioxide that may be found in the turbine flowpath and heat exchangers. This requires the equipment modification to remove CO_2 and the corrosion suppression [18,19]. One more advantage of the STU with oxy-methane superheating is the large knowledge and experience of the mixture combustion in steam at moderate pressures and the effective water condensing from a steam-air mixture [20–22].

Some papers are concerned with the development of the STU with steam overheating in the side-mounted oxy-fuel combustion chambers [23, 24]. These papers review the following power unit heat circuit. The first combustion chamber heats the steam that left the boiler at 600°C temperature and 14 MPa pressure. The combustion chamber heats steam up to 1000°C. After the high-pressure turbine, the second combustion chamber heats steam up to 1250°C. The regenerative system of feeding water heating consists of three high-pressure heaters, a deaerator, two low-pressure heaters and two regenerative coolers of the low-pressure turbine exhaust steam-gas mixture. The 25 MW STU has a net efficiency of 51.7%.

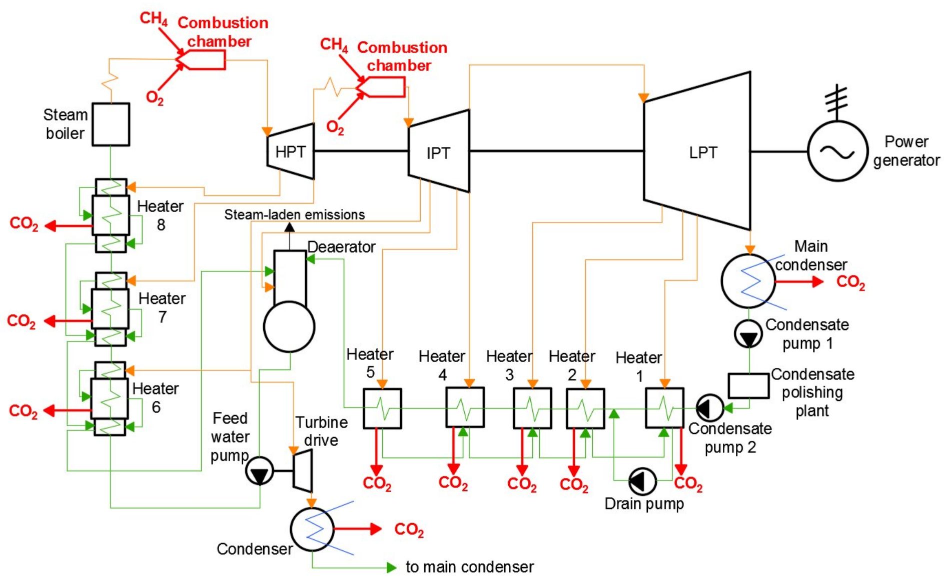
It is worth mentioning that the paper does not assess the TPP efficiency in the range of high supercritical 600–650°C (already established abroad) and ultra-supercritical 700–780°C temperatures. Besides, the prototype facility has a subcritical pressure of 14 MPa, and the efficiency of the secondary reheat is not considered.

This paper discloses the thermodynamic analysis of supercritical and ultra-supercritical steam turbine power plants with side-mounted oxy-methane superheating combustion chambers. The turbine inlet and reheat steam pressure and temperature have been optimized. The investigation includes an analysis of the influence of the side-mounted reheat upon the low-pressure turbine performance.

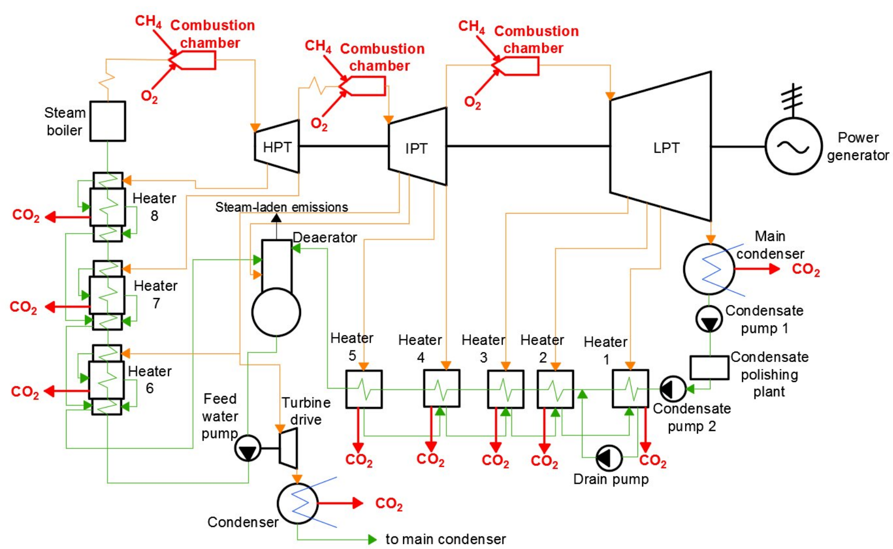
2 Research object

This paper's research object is a steam turbine unit with steam superheating in oxy-methane combustion chambers. The first and second versions of the power plant's scheme have single and double reheat, respectively, operating on an oxy-methane mixture.

Figure 2a presents a heat flow chart of a steam turbine unit with external steam superheating in an oxy-methane combustion chamber and one intermediate superheater. The once-through steam boiler produces supercritical steam and supplies it to the first combustion chamber where the steam is heated up to high supercritical or ultra-supercritical parameters. This combustion chamber is also supplied with preliminary compressed oxygen and methane. The superheated steam expands in the high-pressure turbine and is sequentially heated first by the boiler heating surface and second in the combustion chamber. After the reheat steam expands in the intermediate- and low-pressure turbines, it is sent to the condenser. After the condenser the water pressure is raised in the condensate pump 1, then it is cleaned in the condensate purification plant and its pressure is raised in the condensate pump 2. The STU regeneration system consists of three high-pressure superheaters, a deaerator and five low-pressure superheaters. Carbon dioxide is precipitated from the working fluid in heat exchangers after the steam condensation. The feeding pump is driven by a steam turbine that is supplied with the turbine bleed-



(a) single reheat



(b) double reheat

Figure 2: Cycle arrangements of the steam turbine power plants with external steam superheating in the side-mounted oxy-methane combustion chambers.

ing steam together with the regeneration system, later the used steam is condensed.

The second heat flow chart in Fig. 2b differs from the first one with the third oxy-methane combustion chamber installed between the intermediate and low-pressure turbines. In this combustion chamber, the steam temperature is elevated before the low-pressure turbine inlet.

Two types of air separation units (ASU) were considered for oxygen production: cryogenic for high purity levels of oxygen (>84%) and membrane for low purity levels of oxygen (42%).

Input data for the mathematical modeling are summarized in Table 1.

Table 1: Input data for calculation of the STU cycle arrangements.

Parameter	Unit	Value
Initial pressure	MPa	30
Steam boiler outlet temperature	°C	540
Initial temperature	°C	620–780
Reheat temperature	°C	620–780
The secondary reheat temperature	°C	620–780
Deaerator pressure	MPa	0.7
Steam turbine outlet pressure	kPa	4
Feed water temperature	°C	300
Internal relative efficiency of the HPT/IPT/LPT/turbo feed pump	%	88/91/85/85
Combustion chamber efficiency	%	0.98
Methane pressure at the outlet of the gas reducing station	MPa	0.7
Methane temperature at the outlet of the gas reducing station	°C	15
Oxygen pressure at the outlet of the ASU	MPa	0.1175
Oxygen temperature at the outlet of the ASU	°C	24
Internal relative efficiency of the fuel and oxygen compressors	%	85
Temperature differences in the HP and LP heaters	°C	3
Condensate cooling in the HP heaters	°C	12
Deaerator approach	°C	15
Specific auxiliary electricity consumption except for oxygen production and compression and methane compression	%	3.5

3 Computer simulation model of a steam turbine power plant with an external steam superheating in oxy-methane combustion chambers

The STU models with single (Fig. 3a) and double (Fig. 3b) reheat in the side-mounted combustion chambers were developed with the Aspen Plus software package widely used in industry [27] as well as the ASU model for the investigation of thermal efficiency of a steam turbine power plant with external steam superheating and integrated air separation unit.

The oxy-methane combustion chamber simulation assumed a stoichiometric combustion process. The combustion products are water vapor and carbon dioxide:

$$\mathrm{CH_4} + 2\mathrm{O_2} = 2\mathrm{H_2O} + \mathrm{CO_2} + Q, \tag{1}$$

where Q represents the heating value.

The domestic power consumption for oxygen production is calculated with a developed model of cryogenic low-pressure ASU (Fig. 4). The model evaluates influences of the following parameters upon the ASU performance: oxygen massflow rate (G_{O2}, kg/s), produced oxygen purity ($\mathrm{O_2}$, %), ambient air temperature (t_{atm}, °C), compressor exit pressure ($P_{\mathrm{comp.out}}$, MPa). In this model, the compressor exit pressure corresponds with the assumed upper column pressure. The minimal and maximal oxygen purity degrees are assumed as 84% to 96 in this model according to [25]. The ASU model is verified by the relation of its power consumption and oxygen purity (Fig. 5). The calculated results are compared with the existing facilities' statistical data; the level of results convergence is acceptable.

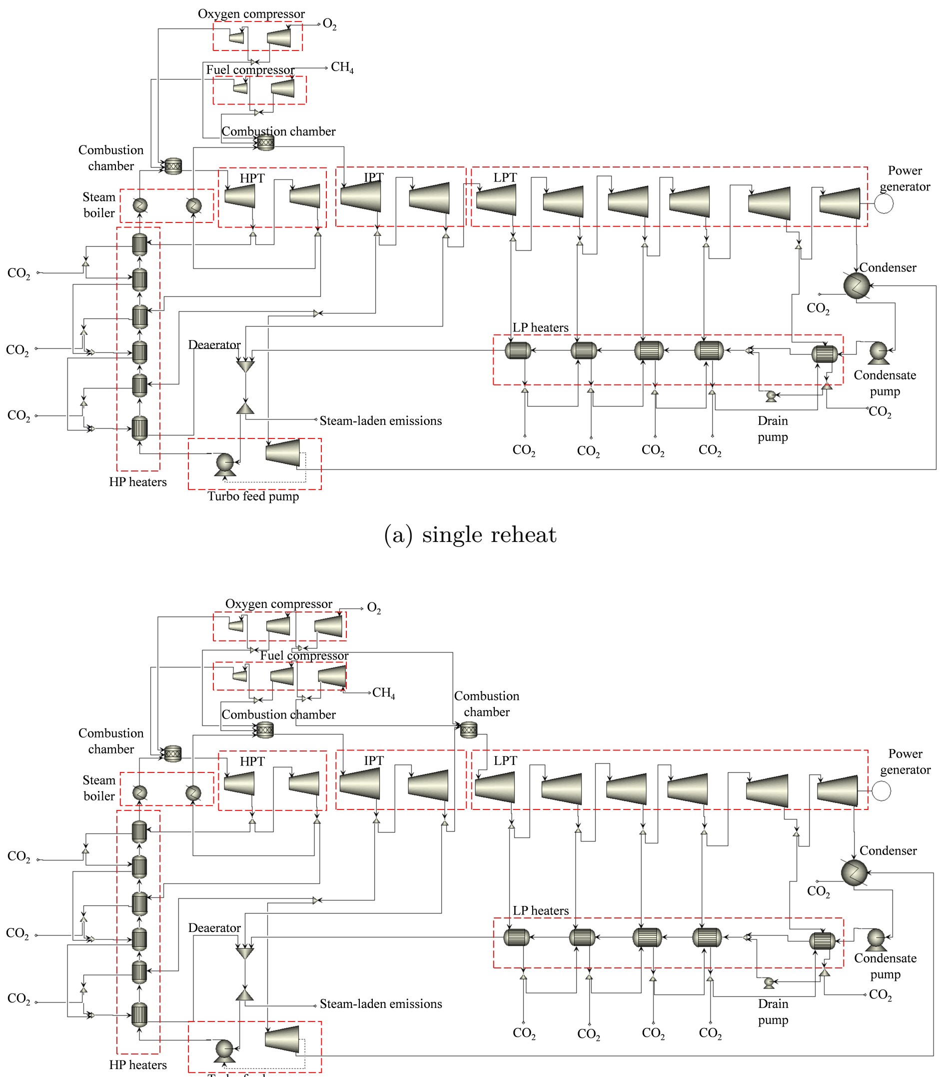
The membrane technology is assumed with a specific power consumption of 580 KWs/kg for the low oxygen purity values [26].

The STU net efficiency with the external reheat combustion chamber was calculated from the following equation

$$\eta_{\mathrm{eff}}^{\mathrm{STU}} = \frac{N_T \eta_{\mathrm{me}} \eta_{\mathrm{gen}} - \dfrac{\left(N_{\mathrm{comp\,O_2}} + N_{\mathrm{comp\,CH_4}} + \dfrac{E_{\mathrm{ASU}}}{1000} G_{\mathrm{O2}}\right)}{(\eta_{\mathrm{me}} \eta_{\mathrm{mot}})}}{(B_{\mathrm{CC}} + B_{\mathrm{SB}})\,\mathrm{LHV}} \times (1 - E_{\mathrm{aux}})\,, \tag{2}$$

where η_{me} is the mechanical efficiency, %; η_{gen} – generator efficiency, %; η_{mot} – motor driver efficiency, %; N_T – turbine generated power, MW;

$N_{comp\,O_2}$ – oxygen compressor power, MW; $N_{\mathrm{comp\,CH_4}}$ – fuel compressor power, MW; B_{CC} – summarized fuel consumption of external combustion chambers, kg/s; B_{SB} – fuel consumption of a steam boiler, kg/s; LHV – low



(a) single reheat

(b) double reheat

Figure 3: Flow chart of the mathematical models of the steam turbine power plants with external steam superheating in the side-mounted oxy-methane combustion chambers.

heating value, MJ/kg; $E_{\rm ASU}$ – specific ASU electricity consumption, determined by Fig. 5, kJ/kg; $G_{\rm O2}$ – oxygen massflow produced by ASU for methane combustion in the combustion chambers, kg/s; $E_{\rm aux}$ – specific auxiliary electricity requirements, %.

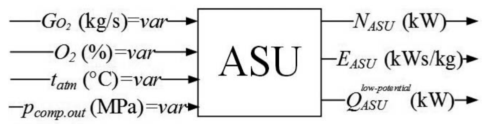


Figure 4: Chart of the ASU parameters and characteristics.

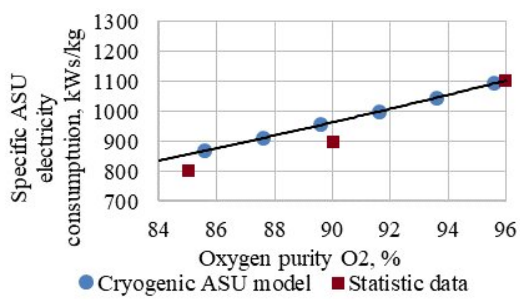


Figure 5: Dependence of the specific ASU electricity consumption on the oxygen purity.

The simulation models were used for thermodynamic analyses of the steam turbine power plant with external steam reheat in the side-mounted oxy-methane combustion chambers.

4 Results and discussion

At the first stage, an investigation of the oxygen content in the oxidizer and the combustion chamber pressure influence on the auxiliary power consumption for the production and supply of 1 kg of oxygen to the combustion chamber was carried out. The following oxidizer content versions were considered:

- air with 21% O_2,
- enriched oxygen with 42% content produced in a membrane ASU,
- enriched oxygen with 84% content produced in a cryogenic ASU,
- enriched oxygen with 96% content produced in a cryogenic ASU.

The calculation results in Fig. 6 show that the smallest power consumption efficiency has the use of air oxidizer with 21% O_2 content and the enriched air oxidizer with 42% O_2 produced in a membrane ASU. On the contrary, the cryogenic ASU shows the smallest production and compression power consumption. Increase of the oxygen purity above 84% does not significantly influence the power consumption. Therefore, the further studies assumed the maximal ASU model purity of 96% that provides minimal impurities in oxidizer and hence in the cycle working fluid.

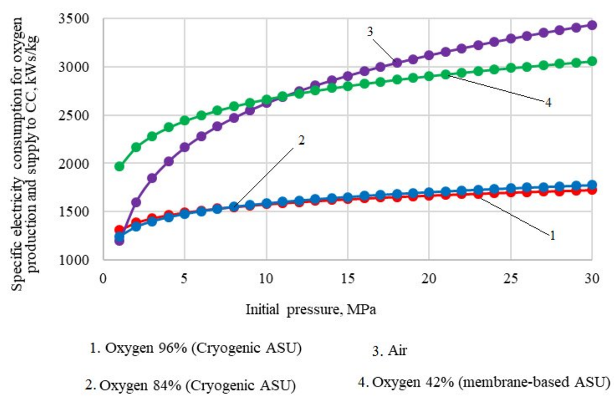
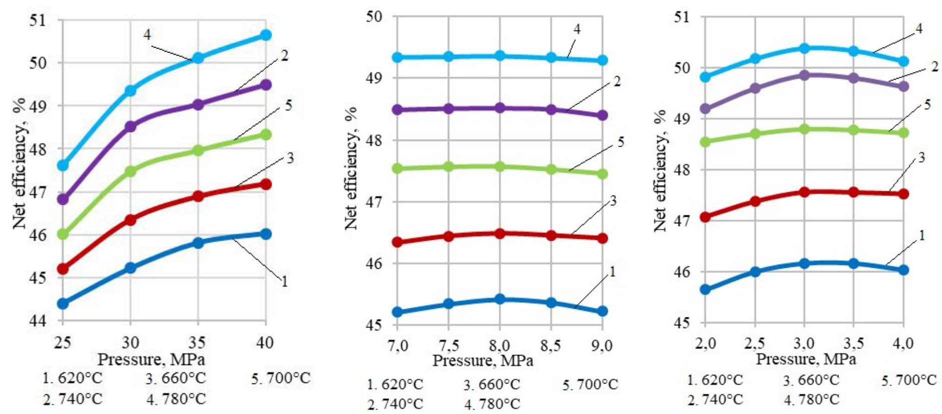


Figure 6: Initial pressure influence on the specific electricity consumption for supplying 1 kg of pure oxygen to the combustion chamber.

The next stage included parametric optimization of the steam turbine inlet pressure and the single and double intermediate reheats in oxy-methane combustion chambers at an initial temperature of 620–780°C. The reheat temperature was assumed equal to the primary steam one. The optimization procedure is the following. At first, the initial pressure influence upon the net efficiency of the STU with a single reheat in the given operating temperature range was investigated (Fig. 7a). Then at the optimal initial pressure the single reheat pressure was optimized (Fig. 7b). Then the secondary reheat pressure influence upon the STU thermal efficiency was studied (Fig. 7c).

The modeling results allow one to conclude that the initial pressure and the STU net efficiency grow in parallel in the initial temperature range of 620–780°C. The pressure increase above 30 MPa may require significantly thicker walls of heat exchangers, boiler heated surfaces and primary steam pipelines. Therefore, further investigation assumes the initial pressure of 30 MPa for the whole initial temperature range.



(a) effect of the initial pressure on the efficiency at single reheat

(b) effect of the reheat pressure on the efficiency at single reheat

(c) effect of the secondary reheat pressure on the efficiency at dual reheat

Figure 7: Working fluid pressure influence on the net efficiency of the steam turbine power plants with single and double reheat in the side-mounted oxy-methane combustion chambers.

The reheat pressure optimization shows that the STU maximum efficiency is reached at the reheat pressure of 8 MPa (27% of the initial pressure) and at the secondary reheat pressure of 3 MPa (10% of the initial pressure). The net efficiency is equal to 45.4–49.4% and 46.1–50.4% for single and dual reheats, respectively, at the initial temperature of 620–780°C.

Compared with the traditional STU where steam is superheated only in the steam boiler, the transition to the external reheat is reasonable at the initial temperatures above 680°C and 650°C for single and dual reheats, respectively (Fig. 8). This tendency is due to the opposite effects of different factors caused by the installation of the side-mounted combustion chambers. On the one side, their application increases the energy losses on fuel and oxidizer preparation. On the other side, the turbine flowpath massflow is increased and the power production is also larger. The external reheat reduces boiler fuel combustion, which reduces the flue gas massflow through the smoke stack. Thus, the single reheat increases the net efficiency by 0.6% at the initial and reheat temperature of 700–780°C, and the dual reheat increases the efficiency by 1.4% at the reheat temperature of 660–780°C.

An important advantage of the external reheat in the side-mounted combustion chambers is the possibility to retain the boiler structure that is the

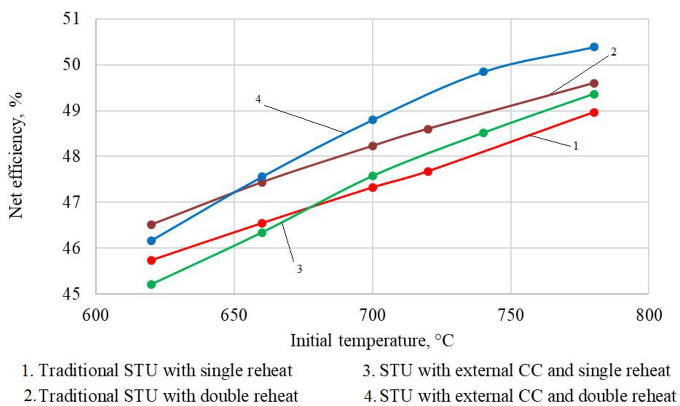
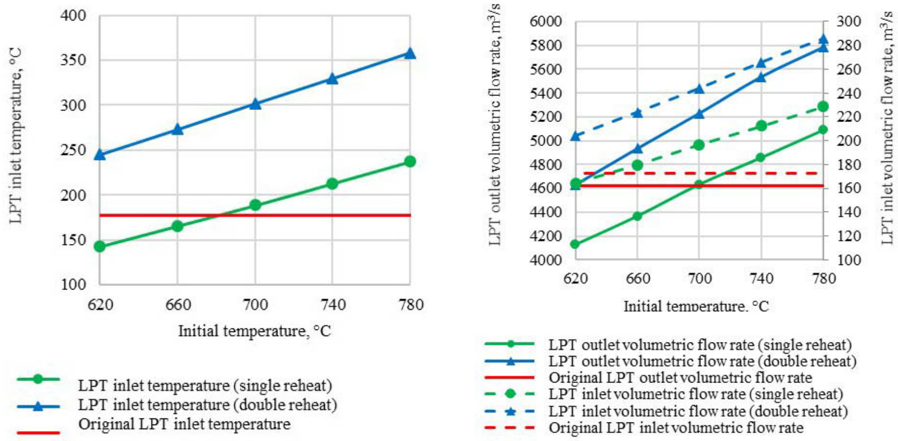


Figure 8: Initial temperature influence on the net efficiency of steam turbine power plants with different types of reheat.

most expensive element of TPP. To analyze the possibility to retain the low-pressure turbine (LPT), which is the most metal-consuming element of steam turbine, the study on the effects of the reheat temperature in side-mounted combustion chambers on volumetric flowrate and temperature of steam at LPT inlet was carried out. The LPT can be retained without significant modernization if mentioned parameters deviation is below 5%. The main results are shown in Fig. 9.

At constant LPT inlet pressure, the LPT inlet temperature may is reduced by 34 and 11°C at the initial temperatures of 620 and 660°C (Fig. 9a). In other cases, the working fluid temperature increases by more than 11°C. The lower LPT inlet temperature at the low initial steam parameters is due to the larger pressure drop in the intermediate pressure turbine caused by the higher reheat pressure.

The study results show that the LPT inlet volumetric flow is influenced by a few factors. First of all, the reheat in external combustion chambers leads to larger massflow rates because the combustion products are mixed with the main flow. Secondly, the reheat in external combustion chambers results in a higher LPT inlet temperature and higher volumetric flow rate at the fixed LPT pressure. Thirdly, the higher reheat increases the carbon dioxide content in the turbine bleeding, which reduces the water vapor partial pressure and increases the bleeding pressure at the fixed level of condensate heating in the heaters. Finally, the bleeding massflow rate grows, which reduces the LPT inlet massflow rate and the related volumetric flow rate.



(a) Low-pressure turbine inlet temperature

(b) Low-pressure turbine inlet and outlet volumetric flow rate

Figure 9: Initial temperature influence on the low-pressure turbine flow parameters for steam turbine power plants with different types of reheat.

The LPT inlet and outlet volumetric flow rate values change less than 5% against the ones in the basic version of LPT at the initial temperature of 660°C and the single reheat (Fig. 9b). These changes are above 5% at other inlet temperatures and both single and double reheats. Thus, it is possible to retain the LPT structure without modernization for the STU only with an initial temperature of 660°C and a single reheat.

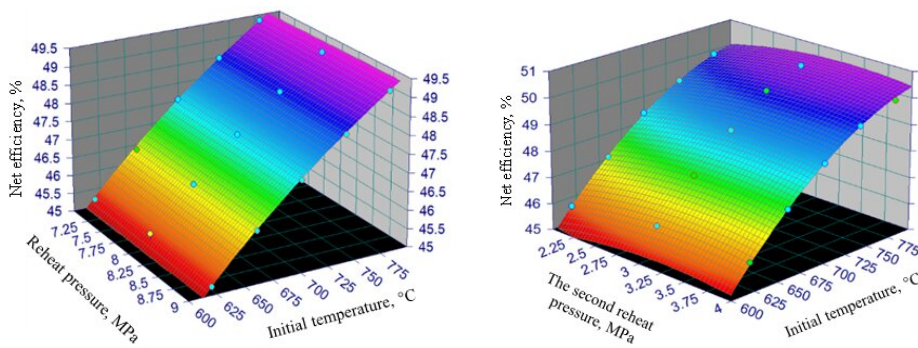
The correlations in equations for the single and dual reheat were developed based on the results of calculation study of the effects of initial and reheat parameters on the net efficiency of the STU with the side-mounted combustion chambers:

$$\eta_{\mathrm{eff}}^{\mathrm{STU(1reheat)}} = 65.17 - 0.013 P_{\mathrm{reheat}_1} - \frac{12280}{t_0}\,, \tag{3}$$

$$\begin{aligned} \eta_{\mathrm{eff}}^{\mathrm{STP(2reheat)}} &= -16.67 + 2.177 P_{\mathrm{reheat}_2} - 0.329 P_{\mathrm{reheat}_2}^2 + \\ &\quad + 0.151 t_0 - 8.914 \times 10^{-5} t_0^2\,, \end{aligned} \tag{4}$$

where P_{reheat_1} is a pressure of the first working fluid reheat, MPa; P_{reheat_2} – a pressure of the secondary reheat, MPa; t_0 – an initial temperature, °C.

Figure 10 presents the dependencies of the STU net efficiency function on the oxy-methane combustion chambers' initial temperature and pressure.



(a) single reheat (b) double reheat

Figure 10: Main parameters influence on the net efficiency of the steam turbine power plant.

5 Conclusions

1. The cycle arrangements and mathematical models of steam turbine units with external oxy-methane combustion chambers with single and double reheats were developed.

2. To achieve high net efficiency of steam turbine unit with the external reheats in oxy-methane combustion chambers the oxidizer must be produced by cryogenic air separation units. The oxygen content should be above 96% to reduce the working fluid impurities content.

3. Single reheat provides maximum net efficiency of 49.4% at the working fluid initial temperature and pressure equal to 780°C and 30 MPa, and at the intermediate reheat pressure of 8 MPa. The maximum net efficiency of 50.4% is reached for the STU power plant with dual reheat at the working fluid initial temperature and pressure of 780°C and 30 MPa, and at the first and secondary reheat pressures of 8 and 3 MPa, respectively.

4. Compared with the traditional steam reheat technology the working fluid reheat in one or two external oxy-methane combustion chambers allows the STU thermal efficiency increase by 0.6% and 1.4%, respectively, at the initial temperature of 700–780°C and 660–780°C.

5. A STU modification by the installation of primary and intermediate steam reheats up to 660°C in the side-mounted combustion chambers

does not require significant low-pressure turbine modification because the temperature and pressure drops do not change substantially and the volumetric flow changes are below 5%.

6. On the basis of thermodynamic analysis results, the correlations of the net efficiency dependencies of the STU with single and double reheat by methane-oxygen combustion chambers from the initial and reheat parameters were developed.

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