

COMPARATIVE ANALYSIS OF THERMODYNAMIC CYCLES OF SELECTED NUCLEAR SHIP POWER PLANTS WITH HIGH- TEMPERATURE HELIUM- COOLED NUCLEAR REACTOR

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ABSTRACT

This paper presents a comparative analysis of thermodynamic cycles of two ship power plant systems with a high-temperature helium- cooled nuclear reactor. The first of them is a gas system with recuperator, in which classical gas chamber is substituted for a HTGR reactor (High Temperature Gas-cooled Reactor). The second of the considered cycles is a combined gas-steam system where working medium flux from gas turbine outlet is directed into waste heat boiler and its heat is utilized for production of superheated steam to drive steam turbine. Preliminary calculations of the combine cycles showed that it is necessary to expand the system by adding to its steam part an inter-stage overheat for secondary steam, owing to that a required degree of steam dryness at outlet from the turbine can be reached, ensuring its correct operational conditions. The analyzed power systems were compared to each other with regard to efficiency of their thermodynamic cycles. Also, efficiency of particular cycles were subjected to optimization in respect to such parameters as: working gas temperature at outlet from reactor in gas system as well as steam pressure at outlet from waste heat boiler and partition pressure in steam part of combined system. Advantages of nuclear power plants compared with the classical power systems dominating currently in sea transport were also discussed.

Keywords: nuclear ship power plant, nuclear reactor, HTGR, gas turbine, steam turbine, combined cycle

INTRODUCTION

Progressing usage of fossil fuels resources and more and more restrictive requirements concerning mitigation of emission of greenhouse gases lead to an increasing interest to alternative propulsion systems for maritime transport purposes. The commonly applied conventional propulsion systems with self-ignition engines or oil firing boilers are emission sources of large amounts of carbon dioxide, sulphur oxides and nitrogen. Their exhaust gases pollute air both in coastal and offshore areas [4, 5, 6].

Nuclear power plants [14] which are successfully used for propulsion of both naval and civilian ships [2] may serve as an alternative for the conventional systems.

One of the types of reactors applicable to nuclear power plants is the high-temperature gas-cooled reactor (HTGR) [9]. It is characteristic of a high temperature at core outlet that makes it possible to use the cooling medium for both driving gas turbine [10] and carrying out various chemical processes such as hydrogen production or crude oil refining. Subsequent advantages of the HTGR reactor is its safety and a high degree of fuel burnup. It is ensured due to a high thermal capacity of graphite which is commonly used as a moderator which maintains structural stability even in high temperature conditions. Moreover, waste heat from nuclear power plant can be used for desalting sea water, heating or producing superheated steam in gas-steam combine power system, namely in the so called waste heat boiler [15].

Nuclear power plants allow to generate power in large amounts. They are characteristic of a high reliability and lowered fuel supply frequency that results in increasing operational range of ships and becoming independent of tankers. The advantages caused that, beginning from the 1950s, nuclear power plants found their wide application [6, 8, 12] to propulsion of icebreakers intended for the operating in polar regions where they perform well in extreme operational conditions delivering necessary supply of food, fuel, equipment etc [1, 8, 11, 13]. In present, nuclear propulsion is widely used in the navy and less often in civilian ships [5,6].

ANALYZED CYCLES OF NUCLEAR POWER PLANTS

The performed calculations were aimed at comparison of efficiency of two nuclear - gas cycles with high- temperature helium-cooled reactor. Owing to its favourable thermal properties, helium performs very well as a cooling medium for the reactor and simultaneously as a working medium in the gas turbine cycle. Thermal conductivity coefficient of helium is five times greater than that of air which commonly serves as a working medium in turbine cycles of combustion engines [3]. Moreover, helium, being a perfect gas, does not undergo ionization, hence it is more friendly to the environment than other cooling media used for nuclear reactors [9].

The first of the considered systems was the gas power system with recuperator (Fig. 1). As opposed to the classical turbine cycle of combustion engine, a high- temperature reactor HTGR was substituted for combustion chamber. And, in view of the high temperature (934 K) of working medium at outlet from gas turbine, a recuperator intended for the heating of working medium before reactor, was applied. As a result, amount of heat wasted in the cycle was lowered and this way the system's effectiveness was improved.

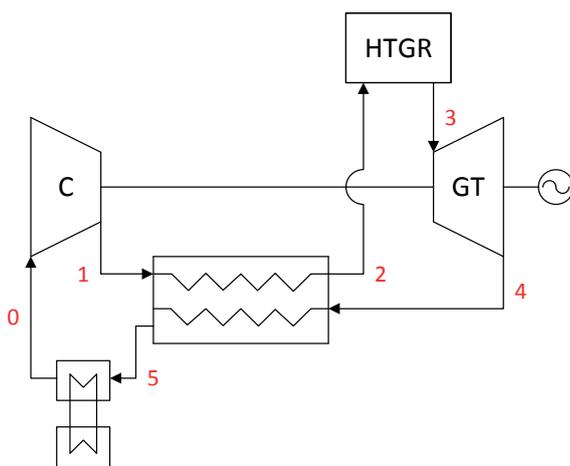


Fig. 1. The gas power system fitted with the high-temperature reactor HTGR and gas recuperator: C – compressor, HTGR – high-temperature reactor, GT – gas turbine

The cycle's parameters were so selected as to suit requirements of the reactor. As assumed, the pressure of helium cooling the reactor, p_2 , is equal to 7 MPa. Application of a lower pressure could cause a significant rise in specific volume of working gas, that would detrimentally influence dimensions and weight of the whole installation. The desired level of cooling medium pressure was reached by appropriate selection of the compression ratio π_κ of the compressor by assuming values of the temperature T_1 and pressure p_1 of working gas at inlet to the compressor equal to 299 K and 2,8 MPa, respectively. Under the so selected parameters the applied high-temperature reactor heats working medium up to the temperature T_3 of 1273 K [4]. The gas temperature value at outlet from the reactor was determined on the basis of predicted amount of heat transferred to cooling medium as a result of nuclear reaction in the reactor. Under assumption that working medium in gas turbine expands up to the pressure, a little greater than that at inlet to the compressor, as well as by taking into account pressure losses in the recuperator and cooler, the value of the expansion ratio in the turbine was taken equal to 2,5. For simplification it was assumed that the parameters which characterize working medium, such as the specific heat c_p or adiabate exponent κ , are constant within the considered range of pressure and temperature changes. For the working medium such as helium the specific heat $c_p = 5,19$ kJ/kgK, and adiabate exponent $\kappa = 1,66$ was assumed.

Another analyzed power system was the combine gas-steam cycle shown in Fig. 2 [7]. In such systems working medium from gas turbine outlet is directed to waste heat boiler in which the heat is used to produce superheated steam supplying steam turbine. Such usage of waste heat of the gas cycle makes it possible to gain an additional portion of power on steam turbine shaft, that significantly increases efficiency of the whole system. In relation to the first of the considered cycles, the working medium parameters at inlet to and outlet from the compressor and turbine in the gas part were maintained without any change. In the combined system the waste heat boiler which produces superheated steam to feed steam turbine is substituted for the recuperator. The steam temperature at outlet from the boiler, t_A , was assumed on the basis of the constant value of difference between live steam temperature and helium temperature at inlet to waste heat boiler ($\Delta t_2 = 30^\circ\text{C}$). Fig. 3 shows a diagram of temperatures in the waste heat boiler. The live steam pressure at outlet from the waste heat boiler was subjected to optimization to make effectiveness of the considered system as good as possible. In selecting its value, a dryness degree of steam at outlet from the steam turbine (which cannot be lower than 0,87 in the final point of expansion) played a similarly significant role as the effectiveness of the cycle. The final pressure in condenser, p_B , was assumed equal to 5 kPa. Also, a minimum temperature difference in the beginning of vaporization process in the waste heat boiler, the so called temperature threshold (Fig. 3), was assumed equal to 8°C .

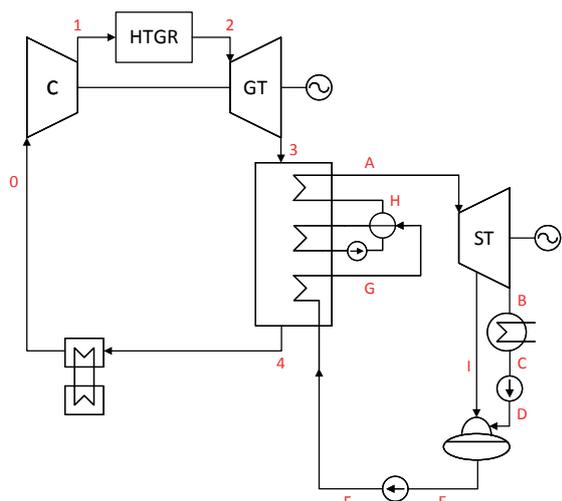


Fig. 2. The combined gas-steam power system fitted with the high-temperature reactor HTGR: C – compressor, HTGR – high-temperature reactor, GT – gas turbine, ST – steam turbine

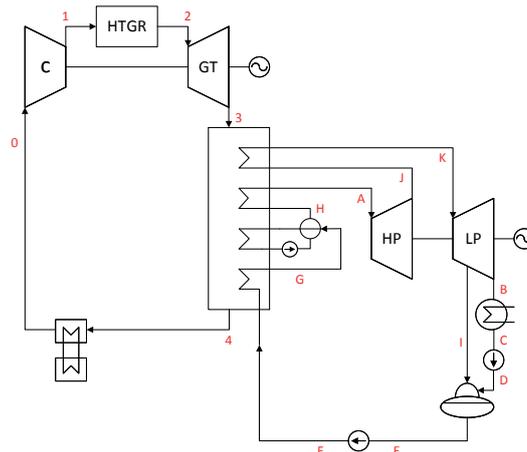


Fig. 4. The combined gas-steam power system fitted with the high-temperature reactor HTGR and inter-stage overheating: C – compressor, HTGR – high-temperature reactor, GT – gas turbine, HP – high-pressure part of steam turbine, LP – low-pressure part of steam turbine

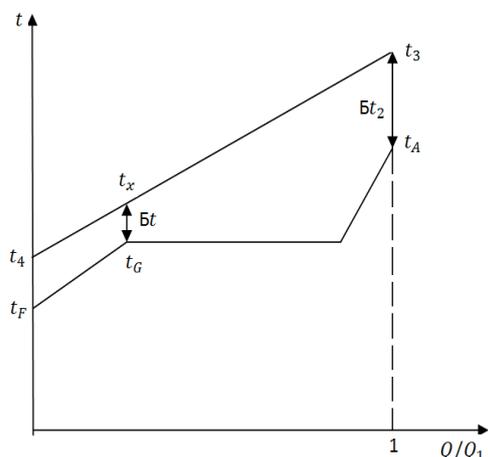


Fig. 3. Diagram of temperatures in waste heat boiler

After performing the preliminary thermodynamic calculations for the combined system its steam part was modified (Fig. 4). In view of a low dryness degree of steam at outlet from the turbine, the system was expanded by adding an inter-stage overheat dividing this way the turbine into two parts: high-pressure (HP) and low-pressure (LP) part. In the so modified system the secondary steam is directed from outlet of HP part into the overheat to be heated there up to a temperature close to that of live steam at outlet from the waste heat boiler. Such operation makes it possible to decrease steam wetness at outlet from LP part and reach consequently a significant improvement in efficiency of the considered combined system.

RESULTS OF THE ANALYSIS

Based on the assumed values of the working medium parameters, values of pressures, temperatures and enthalpies were determined in characteristic points of the considered cycles. In each case, the gas turbine output power was assumed equal to 50 MW. The calculation results for the particular systems are presented in Tab. 1, 2 and 3.

Tab. 1. Thermodynamic parameters of gas cycle with HTGR reactor and gas recuperator

| Parameters of gas cycle with HTGR reactor and gas recuperator | | | |
|---|----|--------|-------|
| Gas turbine output power | Ne | 50 | MW |
| Working medium pressure at inlet to compressor | p0 | 2,8 | MPa |
| Working medium temperature at inlet to compressor | T0 | 299,2 | K |
| Working medium enthalpy at inlet to compressor | i0 | 1552,6 | kJ/kg |
| Working medium pressure at outlet from compressor | p1 | 7 | MPa |
| Working medium temperature at outlet from compressor | T1 | 453,8 | K |
| Working medium enthalpy at outlet from compressor | i1 | 2355,4 | kJ/kg |
| Pressure at inlet to HTGR reactor | p2 | 7 | MPa |
| Working medium temperature at inlet to HTGR reactor | T2 | 837,9 | K |
| Working medium enthalpy at inlet to HTGR reactor | i2 | 4348,9 | kJ/kg |
| Working medium pressure behind HTGR reactor | p3 | 6,8 | MPa |
| Working medium temperature behind HTGR reactor | T3 | 1273 | K |

| Parameters of gas cycle with HTGR reactor and gas recuperator | | | |
|---|----|--------|-------|
| Working medium enthalpy behind HTGR reactor | i3 | 6607,2 | kJ/kg |
| Working medium temperature at outlet from gas turbine | T4 | 954 | K |
| Working medium pressure at outlet from gas turbine | p4 | 3 | MPa |
| Working medium enthalpy at outlet from gas turbine | i4 | 4953,5 | kJ/kg |

Tab. 2. Thermodynamic parameters of combined gas-steam cycle with HTGR reactor without inter-stage overheating

| Parameters of combined gas-steam cycle with HTGR reactor without inter-stage overheating | | | |
|--|----|--------|-------|
| Gas turbine output power | Ne | 50 | MW |
| Working medium pressure at inlet to compressor | p0 | 2,8 | MPa |
| Working medium temperature at inlet to compressor | T0 | 299,2 | K |
| Working medium enthalpy at inlet to compressor | i0 | 1552,6 | kJ/kg |
| Working medium pressure at outlet from compressor | p1 | 7 | MPa |
| Working medium temperature at outlet from compressor | T1 | 453,8 | K |
| Working medium enthalpy at outlet from compressor | i1 | 2355,4 | kJ/kg |
| Working medium pressure behind HTGR reactor | p2 | 6,8 | MPa |
| Working medium temperature behind HTGR reactor | T2 | 1273 | K |
| Working medium enthalpy behind HTGR reactor | i2 | 3648,2 | kJ/kg |
| Working medium temperature at outlet from gas turbine | T3 | 954 | K |
| Working medium pressure at outlet from gas turbine | p3 | 3 | MPa |
| Working medium enthalpy at outlet from gas turbine | i3 | 4953,5 | kJ/kg |
| Working medium temperature behind waste heat boiler | T4 | 410 | K |
| Working medium pressure behind waste heat boiler | p4 | 3 | MPa |
| Working medium enthalpy behind waste heat boiler | i4 | 2128,4 | kJ/kg |
| Live steam pressure | pA | 22 | MPa |
| Live steam temperature | tA | 914,3 | K |
| Live steam enthalpy | iA | 3636,7 | kJ/kg |
| Final pressure | pB | 5 | kPa |
| Steam temperature at outlet from steam turbine | tB | 305,9 | K |
| Steam enthalpy at outlet from steam turbine | iB | 2168 | kJ/kg |

| Parameters of combined gas-steam cycle with HTGR reactor without inter-stage overheating | | | |
|--|----|-------|-------|
| Water pressure behind degasifier | pE | 23 | kPa |
| Water temperature behind degasifier | tE | 335,9 | K |
| Water enthalpy behind degasifier | iE | 263,5 | kJ/kg |

Tab. 3. Thermodynamic parameters of combined gas-steam cycle with HTGR reactor and inter-stage overheating

| Parameters of combined gas-steam cycle with HTGR reactor and inter-stage overheating | | | |
|--|----|--------|-------|
| Gas turbine output power | Ne | 50 | MW |
| Working medium pressure at inlet to compressor | p0 | 2,8 | MPa |
| Working medium temperature at inlet to compressor | T0 | 299,2 | K |
| Working medium enthalpy at inlet to compressor | i0 | 1552,6 | kJ/kg |
| Working medium pressure at outlet from compressor | p1 | 7 | MPa |
| Working medium temperature at outlet from compressor | T1 | 453,8 | K |
| Working medium enthalpy at outlet from compressor | i1 | 2355,4 | kJ/kg |
| Working medium pressure behind HTGR reactor | p2 | 6,8 | MPa |
| Working medium temperature behind HTGR reactor | T2 | 1273 | K |
| Working medium enthalpy behind HTGR reactor | i2 | 6607,6 | kJ/kg |
| Working medium temperature at outlet from gas turbine | T3 | 954 | K |
| Working medium pressure at outlet from gas turbine | p3 | 3 | MPa |
| Working medium enthalpy at outlet from gas turbine | i3 | 4953,5 | kJ/kg |
| Working medium temperature behind waste heat boiler | T4 | 334,7 | K |
| Working medium pressure behind waste heat boiler | p4 | 3 | MPa |
| Working medium enthalpy behind waste heat boiler | i4 | 1737,3 | kJ/kg |
| Live steam pressure | pA | 22 | MPa |
| Live steam temperature | tA | 914,3 | K |
| Live steam enthalpy | iA | 3636,7 | kJ/kg |
| Steam pressure at outlet from HP part of steam turbine (partition pressure) | pJ | 1,76 | MPa |
| Steam enthalpy at outlet from HP part of steam turbine | iJ | 2964,8 | kJ/kg |
| Steam pressure at inlet to LP part of steam turbine | pK | 1,67 | MPa |
| Steam temperature at inlet to LP part of steam turbine | tK | 914,3 | K |

| Parameters of combined gas-steam cycle with HTGR reactor and inter-stage overhear | | | |
|---|----|--------|-------|
| Steam entalphy at inlet to LP part of steam turbine | iK | 3786,3 | kJ/kg |
| Final pressure | pB | 5 | kPa |
| Steam temperature at outlet from LP part of steam turbine | tB | 305,9 | K |
| Steam entalphy at outlet from LP part of steam turbine | iB | 2330 | kJ/kg |
| Water pressure behind degasifier | pE | 23 | kPa |
| Water temperature behind degasifier | tE | 335,9 | K |
| Water entalphy behind degasifier | iE | 263,5 | kJ/kg |

Efficiency of the gas cycle with recuperator was determined form the following formula:

$$\eta_{GT} = \frac{N_T - N_C}{Q_d} = \frac{(i_3 - i_4) - (i_1 - i_0)}{i_3 - i_2}$$

where:

N_T – gas turbine output power

N_C – compressor output power

Q_d – heat flux transferred to cooling medium as a result of nuclear reaction.

Optimization of the efficiency was made in respect to the working medium temperature behind the reactor, T_3 (Fig. 5). Taking into account capability of HTGR reactor, value of T_3 temperature was taken within the range from 1073 to 1273 K [4].

The highest efficiency of 41,7% was obtained for the highest temperature at outlet from the reactor. For lower temperatures, the efficiency of the gas cycle with recuperation varies within the range from 35,5% to 41%.

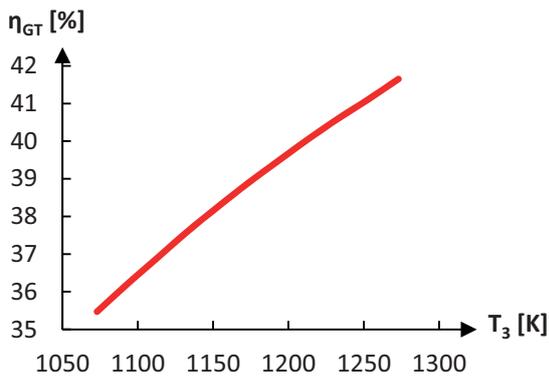


Fig. 5. Diagram of relation between efficiency of the gas cycle with recuperation and working medium temperature at outlet from HTGR reactor

The efficiency of the combined gas-steam cycle without inter-stage overhear was determined from the following formula:

$$\eta_{GS} = \eta_{GT} \left(\frac{N_{ST}}{N_{GT}} + 1 \right)$$

$$N_{ST} = m_0(i_A - i_1) + (m_0 - m_{SE})(i_1 - i_B)$$

$$\eta_{GT} = \frac{N_{GT} - N_C}{Q_d} = \frac{(i_2 - i_3) - (i_1 - i_0)}{i_2 - i_1}$$

where:

η_{GS} – efficiency of combined gas-steam cycle

η_{GT} – efficiency of gas part of the system

N_{ST} – steam turbine output power

N_{GT} – gas turbine output power

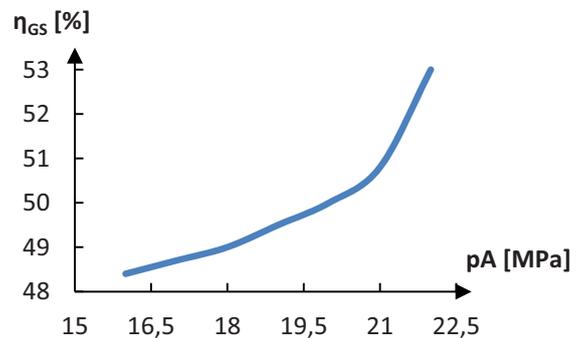
m_0 – live steam mass flux

m_{SE} – mass flux of steam transferred from regenerating extraction to degasifier

N_C – compressor output power

Q_d – heat flux transferred to cooling medium due to nuclear reaction

The combined cycle efficiency was optimized in respect to live steam pressure at outlet from waste heat boiler (Fig. 6).



Rys. 6. Relation between the efficiency of combined gas-steam cycle with HTGR reactor and the live steam pressure (p_A) at outlet from waste heat boiler

Steam dryness degree at outlet from the turbine was also checked. It was assumed that the steam dryness degree should not be lower than 0,87 in order to maintain correct conditions for turbine operation. Value of that factor depends in large measure on an assumed value of live steam pressure. For the assumed range of steam pressure at outlet from waste heat boiler within limits from 16 to 22 MPa it is clearly visible (Fig. 7) that moisture content in working medium increases along with live steam pressure increasing, that unfavourably affects turbine operational conditions. For the preliminarily assumed cycle parameters, any pressure value within the considered range did not allow to reach the desired value of steam dryness degree.

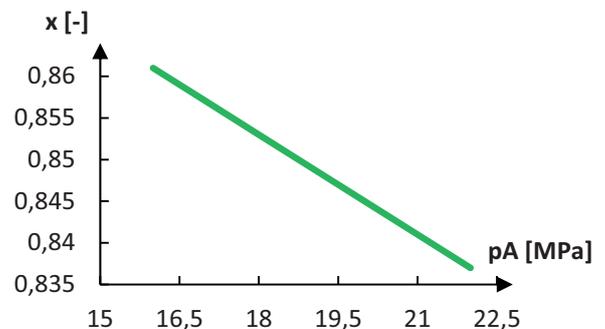


Fig. 7. Relation between the steam dryness degree at outlet from turbine and the live steam pressure (p_A)

The efficiency of the combined system with inter-stage overheat was determined on the basis of the following formulae:

$$\eta_{GS} = \eta_{GT} \left(\frac{N_{ST}}{N_{GT}} + 1 \right)$$

$$N_{ST} = m_0(i_A - i_J) + m_0(i_J - i_I) + (m_0 - m_{SE})(i_I - i_B)$$

$$\eta_{GT} = \frac{N_{GT} - N_C}{Q_d} = \frac{(i_2 - i_3) - (i_1 - i_0)}{i_2 - i_1}$$

The system was optimized in respect to value of the live steam pressure p_A at outlet from waste heat boiler as well as the partition pressure p_J (Fig. 8 and 9). As mentioned above, the introduction of inter-stage overheat makes it possible to apply higher live steam pressure values, however, for purposes of the next comparative analysis of the considered systems, the same values of the principle parameters of the considered cycles, including a. o. live steam pressure range, were assumed.

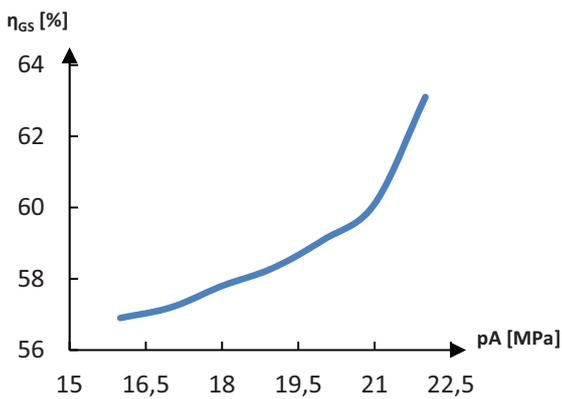


Fig. 8. Relation between the efficiency of combined gas-steam cycle with inter-stage overheat and the live steam pressure (p_A)

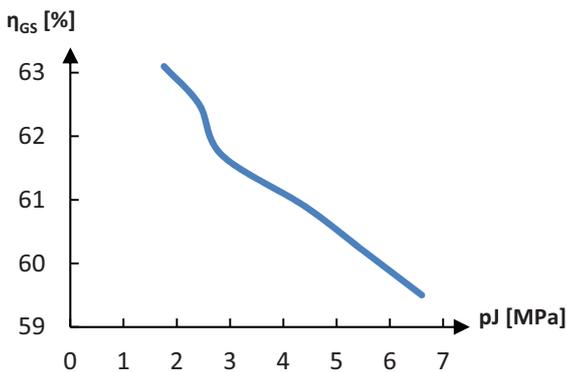


Fig. 9. Relation between the efficiency of combined gas-steam cycle with inter-stage overheat and the partition pressure (p_J)

Like in the case of the combined cycle without inter-stage overheat, the efficiency of the whole system increases along with live steam pressure increasing. For the assumed live steam pressure range the efficiency varies within the limits

from 56,9% to 63,1%. The highest efficiency was obtained for the pressure $p_A = 22$ MPa.

The performed optimization of the partition pressure showed that the cycle efficiency decreases along with pressure increasing at outlet from HP part of the turbine. The optimum value of that parameter was assumed equal to 1,76 MPa which is equivalent to 8% value of the steam pressure p_A at outlet from waste heat boiler.

SUMMARY

The performed comparative analysis of thermodynamic cycles of the selected ship nuclear power systems with high-temperature helium - cooled nuclear reactor showed that:

- from the point of view of cycle efficiency and amount of generated power, the using of waste heat for production of superheated steam to drive steam turbine is more favourable than the application of gas recuperator ;
- the rise in working medium temperature at outlet from the core affects favourably thermodynamic cycle efficiency, that explains growing interest for VHTR reactors;
- in the case of the combined gas-steam power system a level of live steam pressure and partition pressure is strictly dependent upon steam dryness degree at outlet from the turbine;
- to reach a desired value of steam dryness degree in the combined cycle, it is necessary to apply inter-stage overheat for secondary steam.

Widespread of application of nuclear propulsion systems with high-temperature reactors would allow to reduce emission of noxious gases in coastal and offshore areas and diversify fuels currently used in sea transport . Along with growing popularity of nuclear power plants their unit cost of production would be dropping and they would become, as a result of growing public awareness, strongly competitive for the commonly used classical power plants.

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