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Effectiveness analysis of a binary ORC power plant with zeotropic organic fluid

Sławomir Wiśniewski[a*], Michał Bańkowski[a]

[a]West Pomeranian University of Technology, Faculty of Mechanical Engineering and Mechatronics, 19 Piastów Avenue, Szczecin
*Corresponding author email: slawomir.wisniewski@zut.edu.pl

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Abstract

A review of the available literature shows that analyses of organic Rankine cycle systems with a zeotropic mixture working medium practically concern single-circuit systems. In these works, it has been shown that the standing of zeotropic mixtures in organic Rankine cycle systems makes it possible to achieve higher power and efficiency compared to organic Rankine cycle systems with pure fluids. In this article, the authors present an analysis of the efficiency of a two-circuit organic Rankine cycle (binary) power plant with a zeotropic mixture in the upper cycle of this power plant. The proposed binary power plant system uses a zeotropic mixture circulating medium in the upper organic Rankine cycle circuit, while the lower circuit uses a homogeneous low-boiling medium. The results of this analysis showed that with properly selected parameters of the binary power plant system, i.e. with appropriate selection of the pressure during the evaporation transformation in the upper and lower circuits, the power obtained in it is higher than for a single-circuit power plant in the same temperature range (for the same heat source and the same condensing temperature). The increase in the power of the binary power plant system was achieved by using the heat contained in the water stream to preheat the medium in the bottom circuit. For example, for the binary organic Rankine cycle power plant with R413A refrigerant in the upper circuit, the generated power is 17.8 kWe, which is 20% higher than for a single-circuit power plant (for the reference power plant, the power is 14.8 kWe).

Keywords: C cycle; Binary power plant; Top cycle; Lower cycle; Zeotropic mixtures

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1. Introduction

The drastic increase in energy prices due to the current geopolitical situation, the costs associated with CO_2 emission rights and the need to protect the environment give a strong impetus to activities directed at the search for new installations allowing electricity to be generated from sources other than fossil fuels. Fuel cells using hydrogen as fuel are an example of this type of installation. Examples of research on improving the efficiency of proton exchange membrane (PEM) type fuel cells can be found in the works [1,2]. During the operation of fuel cells, waste heat is generated, which can be used to power the organic Clausius-Rankine cycle. Power plants of this type allow the management of low- and medium-temperature energy carriers

Nomenclature

h – specific enthalpy of circulating fluid, kJ/kg
$\dot{m}$ – mass flowrate of circulating fluid, kg/s
N – power, kW
$\dot{Q}$ – heat flux, kJ/s
s – entropy, kJ/(kg K)
t, T – temperature, °C, K

Greek symbols
η – efficiency

Subscripts and Superscripts
1, 2*s*, 3, 4*s*, 5, 6 – of the upper cycle
I, II*s*, III, IV*s*, V, VI– of the lower cycle
B – binary cycle
CR – Clausius-Rankine cycle
L – lower cycle
R – reference cycle
U – upper cycle
*w*1, *w*2, *w*3, *w*4 – water supplied from a heat source

(thermal energy), which makes it possible to use waste energy streams to power these systems. Such a solution improves the energy efficiency of individual technological processes, thereby reducing the economy's energy intensity. Due to the low thermal parameters of heat sources in organic Rankine cycle (ORC) power plant systems powered by waste energy streams, low efficiencies are achieved. However, when using waste energy streams, the aim should be to maximise power output. For this reason, the authors in this paper analysed a binary power plant in the sense of a two-cycle power plant, in which the condensation heat from the upper cycle is used to power the lower cycle. This article aims to prove that a binary power plant can obtain greater power than a single-circuit power plant operating in the same temperature range.

In most cases, homogeneous circulating substances are used in ORC power plant cycles, including binary power plants [3]. An important aspect of the use of these substances is its impact on global warming. This impact is determined by the global worming potential (GWP) index. It is worth noting that under current legislation [4], a group of hydrofluorocarbons (HFCs) can be used in part whose GWP index does not exceed the value of 2 500. The use of a fluid with GWP > 2 500 from 2020 is prohibited. The possibility of using a working fluid with very good thermodynamic properties that are safe for use (e.g., non-flammable) but have a GWP index > 2 500 is possible by mixing it with another fluid with a lower GWP so that the resultant mixture index does not exceed the permissible value. For this reason, the analysis included zeotropic mixtures such as R413A, R423A, R426A, R429A, R430A, R435A and R437A. As a result of their analyses, the assumption of an increase in the power obtained in a binary power plant compared to a single-circuit power plant operating in the same temperature range was confirmed. A detailed description of selected zeotropic mixtures is presented in the next section.

1.1. Use of zeotropic mixtures in ORC systems – literature review

Organic Rankine cycle systems have become an area of intensive research as a promising technology for converting low-grade heat into electricity due to the fact that it has a low-cost, simple structure and can utilise various types of heat sources such as solar energy, geothermal energy and waste heat [5].

In the published literature, we find a variety of models to find the best working fluids based on operating conditions [6−8]. In addition, zeotropic mixtures have been proposed as alternative working fluids for ORC systems with homogeneous fluids, as has the use of mixtures in refrigeration systems and heat pumps [9−10].

Compared to homogeneous fluids, zeotropic mixtures are characterised by temperature slippage during the phase transformation process, thus alleviating the temperature profile mismatch in the evaporator and condenser [11]. Chys et al. [12] analysed the use of working fluid mixtures in ORC systems. It has been shown that cycle efficiency can be increased by 16% and power output by almost 20%. Lecompte et al. [13] studied the efficiency of ORCs using different mixtures based on the second law of thermodynamics. They found that the increase in efficiency ranged from 7.1% to 14.2% compared to ORCs using homogeneous working fluids.

It has been shown that the efficiency of the system can be increased by 16%, and power can be improved by nearly 20%. Habka and Ajib [14] performed performance evaluations of several mixtures of zeotropic working fluids for ORC systems. The study proved that mixtures of R438A, R422A and R22M fluids have better performance than homogeneous fluids.

On the other hand, Ochoa et al.[15] conducted an analysis of energy and exergy to evaluate the durability of the overall system and the viability of the zeotropic mixture from an environmental point of view. The results showed that zeotropic mixtures had better performance compared to selected pure fluids in terms of net power and exergy efficiency. Pure fluids had a better energy sustainability index (ESI). Averaging 10%, there is not much difference in this parameter, so the advantages of using zeotropic mixtures as working fluids in this type of system cannot be ignored.

Some work [16] notes the effect of zeotropic fluids on heat transfer through the condenser, indicating that a much larger heat transfer surface area is required for a zeotropic fluid. On the other hand, the work of Blondel et al. [17] points out that it is often pointed out in the literature that there is an increased thermodynamic cycle efficiency in the use of zeotropic mixtures due to improved heat transfer in the hot spring, but the vast majority of these studies are numerical and do not take into account the interactions and system effects occurring in the thermodynamic cycle for zeotropic mixtures (predict biphasic heat transfer coefficients of pure fluids and zeotropic mixtures). The work of Xia et al. [18] indicates that the influence of mixture composition on the performance of the system is different at different positions of the phase transition temperature. In addition, it was found that the use of a zeotropic mixture is not appropriate in all cases.

Analysis of theoretical and practical calculations shows that organic Rankine circulation using zeotropic working fluids has potential for application. However, there are many challenges to be solved in theory and application, such as key design, change of circulating fluid composition, dynamic performance and control strategy, and so on [19].

All the works cited are concerned with the analysis of ORC power plant systems with circulating agents in the form of zeotropic mixtures. The analyses presented in these works have shown that the use of zeotropic mixtures in ORC systems has a positive effect on their operating efficiency. For ORC systems with zeotropic mixtures, power output is increased compared to ORC systems with pure fluids. However, all these analyses concern ORC systems with one cycle (there are no analyses of two-circuit - binary systems). For this reason, this article presents an analysis of a two-circuit ORC power plant - a binary ORC power plant. For this powerhouse, it was assumed that in the upper circuit the working substance is a zeotropic mixture. The operating efficiency of the binary power plant was compared with that of the single-circuit power plant, maintaining the same temperature ranges for both systems. The purpose of the analysis is to demonstrate the impact of replacing a single-circuit ORC power plant with a binary power plant using zeotropic mixtures.

2. Description of the binary power plant system

The paper presents an analysis of the operating efficiency of a binary ORC power plant with a zeotropic low-boiling medium. For the binary ORC power plant, the assumption was made that the upper cycle used a zeotropic organic medium mixture, and the lower cycle used an organic medium from the dry medium group. The upper cycle was thermally coupled to the lower cycle by means of an HE2 (see Fig. 1) condenser-evaporator type exchanger. This coupling allows the condensation heat from the upper cycle to be used to heat, evaporate and superheat the medium in the lower cycle [3]. Due to the use of a circulating medium in the lower cycle from the dry medium group, two variants of the binary power plant were included in the analysis. The first variant of the binary power plant with lower circulation without the use of heat regeneration is shown in Fig. 1.

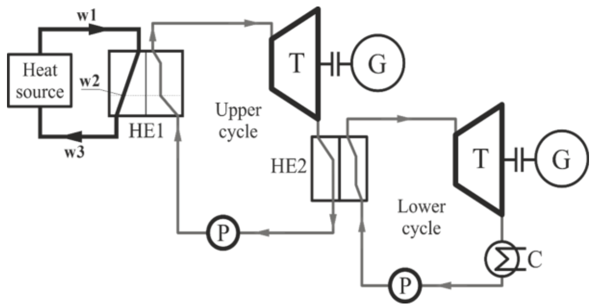

Fig. 1. Schematic of a binary power plant (two-cycle power plant).

The second variant of the binary power plant with a heat exchanger that allows heat recovery from the steam leaving the lower-cycle turbine is shown in Fig. 2. The use of this HE3 exchanger is possible due to the fact that for a dry medium, the steam after expansion in the turbine is still superheated steam with a temperature higher than the condensing temperature in the lower cycle [20,21].

Figure 3 shows a diagram of thermodynamic transformations for the individual cycles of a binary power plant operating according to the scheme shown in Fig. 1 or Fig. 2. In this case, energy from the heat source is transferred to the binary power plant system only in the HE1 exchanger.

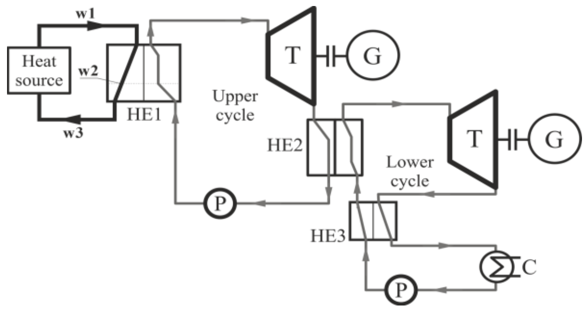

Fig. 2. The scheme of the binary power plant with the applied internal heat exchange in the lower cycle.

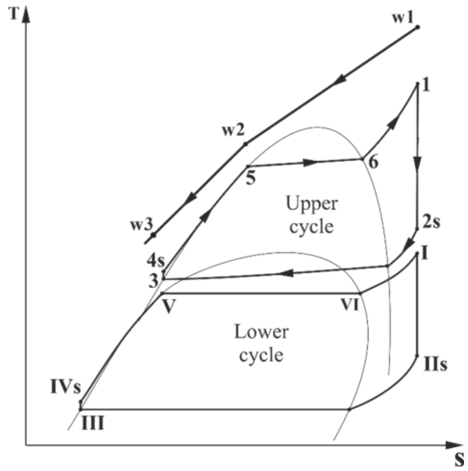

Fig. 3. Thermodynamic transformation diagram for a binary power plant

Analysis of the binary power plant layouts shown in Figs. 1 and 2 revealed that the achievable capacities of these power plants are lower than for the reference power plant (single-cycle power plant) [22,23]. In addition, it was noted that the temperature of the water returning to the heat source, i.e. the temperature of the water downstream of the HE1 heat exchanger, is high at about 90°C. Therefore, the stream of water leaving the HE1 exchanger was used to preheat the working fluid in the lower cycle of the binary power plant. Water from the HE1 exchanger is directed to the HE4 exchanger, where the working medium of the lower cycle is heated up to the saturation state (point V – see Fig. 3). A diagram of the binary power plant operating according to this description is shown in Fig. 4.

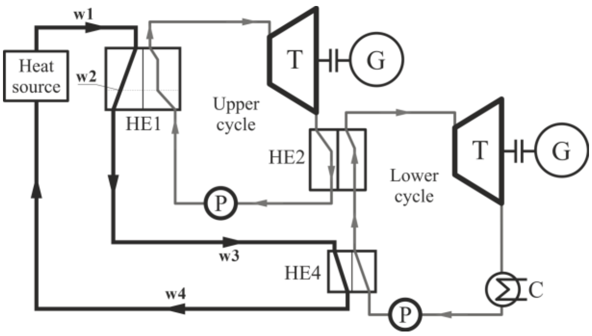

Fig. 4. The scheme of a binary power plant with a pre-heater in the lower cycle.

Figure 5 shows a diagram of thermodynamic transformations for the individual cycles of a binary power plant operating according to the diagram shown in Fig. 4. In this case, energy from the heat source is transferred to the binary power plant system in the HE1 and HE4 exchangers.

Based on previous analyses [24,25], it was assumed that a selection of zeotropic mixture from the 400 series, such as R413A, RTe 423A, R426A, R429A, R435A and R437A, would be analysed in the upper cycle. The basic properties of these zeotropic mixtures are shown in Table 1.

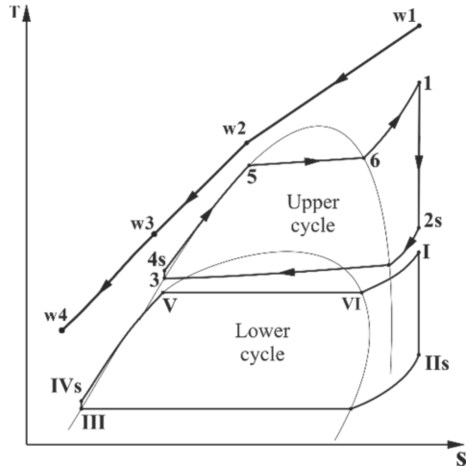

Fig. 5. Thermodynamic transformation diagram for a binary power plant with a pre-heater in the lower cycle.

Table 1. Basic properties for the analyzed zeotropic mixture.

No.	Name of fluid	Type	Composition	Indicator ODP	Indicator GWP_{100}	Density (kg/m^3)	Molecular mass (u)	Critical temperature (°C)	Critical pressure (kPa)	Temp. glide ΔT1 (°C)
1	R-413A	HFC	9±1% C_3F_8 88±2% $C_2H_2F_4$ 3±1% C_4H_{10}	0	2 053	512.53	104	95.335	3 919.8	1.131
2	R-423A	HFC	52.5±1% C_2H_2F 47.5±1% C_3HF_7	0	228	537.18	126	99.136	3 563.0	0.49
3	R-426A	HFC	5.1±1% C_2HF_5 93±1% $C_2H_2F_4$ 1.3±0.2% C_4H_{10} 0.6±0.1% C_5H_{12}	0	1 508	503.13	101.6	98.993	4 018.1	0.48
4	R-429A	HFC	60±1% C_2H_6O 10±1% $C_2H_4F_2$ 30±1% C_4H_{10}	0	139	266.79	50.8	121.95	4 730.0	0.94
5	R-435A	HFC	80±1% C_2H_6O 20±1% $C_2H_4F_2$	0	256	292.02	49	123.06	5 191.2	0.224
6	R-437A	HFC	19.5±0.5% C_2HF_5 78.5±1.5% C_2H_2F 1.4±1% C_4H_{10} 0.6±0.2% C_5H_{12}	0	1 805	517.32	103.7	95.502	4 024.5	1.631
ODP - ozone depletion potential										

The use of a zeotropic medium causes evaporation and condensation transformations in the upper cycle to take place with temperature slippage. An example diagram of the thermodynamic transformations taking place in the cycles of the binary power plant and the temperature distribution for the heat carrier is shown in Fig. 3 or Fig. 5.

In the lower cycle of the binary power plant analysed, it was assumed that the evaporation of the low-boiling medium takes place in the near-critical region, which means that the temperature of this transformation is several K lower than the temperature at the critical point. Previous work has shown that such a solution is advantageous from the point of view of ORC system efficiency [26].

Due to the assumption that the heat source is of a waste energy nature, the authors considered the power of the system as the basic parameter determining its operating efficiency.

In order to evaluate the efficiency of operation of the analyzed binary power plant systems, the obtained calculation results were compared with the power obtained in a reference power plant (single-cycle power plant). In this power plant, the same temperatures of evaporation and superheating of the working medium were assumed for the upper cycle of the binary power plant. However, the condensation temperature of the working medium in the single-cycle power plant is equal to the condensation temperature of the working medium in the lower cycle of the binary power plant. A diagram of the reference (single-cycle) power plant is shown in Fig. 6 (a – the simple ORC system, b – the ORC system with internal heat regeneration).

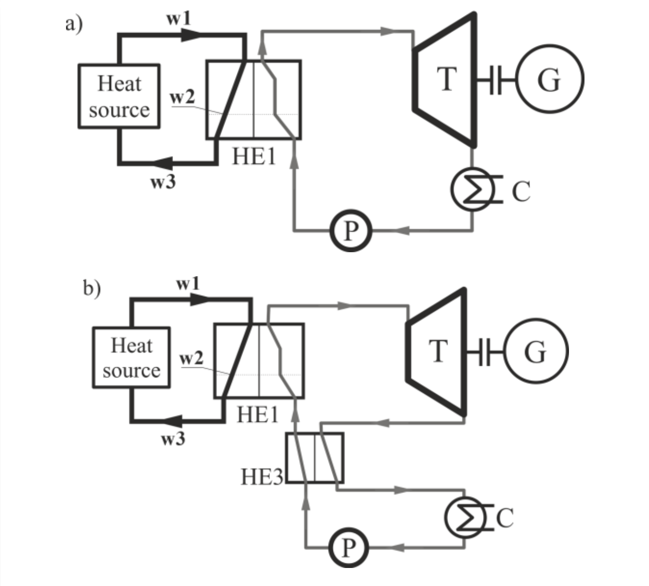

Fig. 6. Single-cycle reference power plant: a) the simple ORC system, b) the ORC system with internal heat regeneration

Figure 7 shows the shape of the saturation curves for the analyzed working fluids in the T–s coordinate system (Temperature–Entropy).

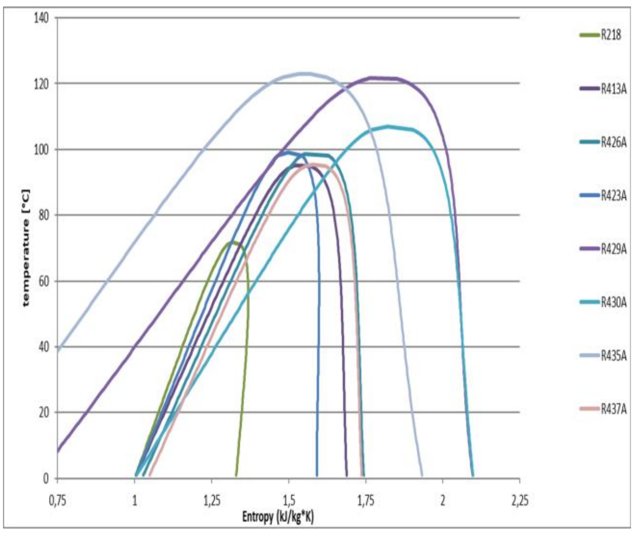

Fig. 7. The shape of the saturation curves for the analyzed working fluids in the T–s coordinate system

3. Methodology of calculations

The detailed methodology for calculating binary power plants can be found, among others, in the works [3,5], and all thermal and caloric parameters of the circulating media and water as the heat carrier were determined using the RefProp 9.1 database of low-boiling media [27]. Selected relationships used in the calculations are presented below.

3.1. Upper cycle

The energy balance equation, ignoring heat losses, for the HE1 exchanger for the superheater and evaporator parts takes the following form:

$$\dot{m}_w c_w (t_{w1} - t_{w2}) = \dot{m}_u (h_1 - h_5), \tag{1a}$$

$$\dot{m}_w (h_{w1} - h_{w2}) = \dot{m}_u (h_1 - h_5). \tag{1b}$$

The mass flow of the working medium in the upper cycle $\dot{m}_u$ was determined by transforming equation (1b):

$$\dot{m}_u = \frac{\dot{m}_w (h_{w1} - h_{w2})}{h_1 - h_5}. \tag{2}$$

The energy balance equation for the HE1 exchanger takes the following form:

$$\dot{Q}_{HE1} = \dot{m}_w c_w (t_{w1} - t_{w3}) = \dot{m}_u (h_1 - h_{4S}). \tag{3}$$

The temperature of the water leaving the HE1 heat exchanger was determined from the transformed relationship (3) as follows:

$$t_{w3} = t_{w1} - \frac{\dot{m}_u (h_1 - h_{4S})}{\dot{m}_w c_w}. \tag{4}$$

3.2. Lower cycle

The energy balance for the HE2 condenser-evaporator exchanger for the binary power plant system shown in Fig. 2 takes the form:

$$\dot{Q}_{HE2} = \dot{m}_U (h_{2s} - h_3) = \dot{m}_L (h_{\mathrm{I}} - h_{\mathrm{IV}s}). \tag{5}$$

The mass flow of the working medium in the lower cycle $\dot{m}_L$ was determined from the above equation:

$$\dot{m}_L = \frac{\dot{m}_U (h_{2s} - h_3)}{h_{\mathrm{I}} - h_{\mathrm{IV}s}}. \tag{6}$$

In the case of the binary power plant system shown in Fig. 4, the energy balance equation for the HE2 condenser-evaporator exchanger takes the form:

$$\dot{Q}_{HE2} = \dot{m}_U (h_{2s} - h_3) = \dot{m}_L (h_{\mathrm{I}} - h_{\mathrm{V}}). \tag{7}$$

The mass flow of the working medium in the lower cycle was determined from the above equation $\dot{m}_L$:

$$\dot{m}_L = \frac{\dot{m}_U (h_{2s} - h_3)}{h_{\mathrm{I}} - h_{\mathrm{V}}}. \tag{8}$$

The medium is heated in the lower cycle for the binary power plant shown in Fig. 4 in the HE4 exchanger using water supplied from the HE1 exchanger.

The temperature of water returning to the heat source was determined from the energy balance equation for the HE4 exchanger:

$$\dot{Q}_{HE4} = \dot{m}_w c_w (t_{w3} - t_{w4}) = \dot{m}_L (h_{\mathrm{V}} - h_{\mathrm{IV}s}), \tag{9a}$$

$$t_{w4} = t_{w3} - \frac{\dot{m}_L(h_{\mathrm{V}} - h_{\mathrm{IV}s})}{\dot{m}_w c_w}. \quad (9b)$$

3.3. Upper cycle of binary power plant

Below are the methods for determining unit values (referred to 1 kg of circulating medium) characterizing the Clausius-Rankine cycle.

The specific isentropic work of the pump (for pumping the low-boiling medium in the upper cycle) is the difference between the specific enthalpies of the medium at the pump outlet and at the pump inlet:

$$l_{U\,com} = h_{4s} - h_3. \quad (10)$$

Taking into account the internal efficiency of the pump η_{ip} = 0.80 and the mechanical efficiency of the pump η_{mp} = 0.99, the unit work required to drive the circulating pump l_{Up} was determined as follows:

$$l_{Up} = \frac{l_{U\,com}}{\eta_{ip}\,\eta_{mp}} = \frac{h_{4s} - h_3}{\eta_{ip}\,\eta_{mp}}. \quad (11)$$

The specific isentropic expansion work for the upper cycle was determined as the difference in specific enthalpy before and after the turbine:

$$l_{Uexp} = h_1 - h_{2s}. \quad (12)$$

Taking into account the internal efficiency of the turbine η_{it} = 0.75 and the mechanical efficiency of the turbine η_{mt} = 0.99, the mechanical unit work of the turbine l_{Ut} was determined as follows:

$$l_{Ut} = \eta_{it}\,\eta_{mt}\,l_{Uexp} = \eta_{it}\,\eta_{mt}(h_1 - h_{2s}). \quad (13)$$

The mechanical power of the upper cycle N_U, taking into account pomp work, was determined as follows:

$$N_U = \dot{m}_U(l_{Ut} - l_{Up}). \quad (14)$$

The electrical power generated in the upper cycle $N_{el\,U}$ was determined by assuming generator efficiency η_g = 0.99 as follows:

$$N_{el\,U} = \eta_g N_U. \quad (15)$$

The heat flux supplied to the upper cycle was determined as follows:

$$\dot{Q}_U = \dot{m}_U(h_1 - h_{4s}). \quad (16)$$

The electrical efficiency of the upper cycle $\eta_{el\,U}$ can be determined from the relationship:

$$\eta_{el\,U} = \frac{N_{el\,U}}{\dot{Q}_U}. \quad (17)$$

3.4. Lower cycle of binary power plant

The specific isentropic work of the pump (for pumping the low-boiling medium in the lower cycle) is the difference between the specific enthalpies of the medium at the pump outlet and at the pump inlet:

$$l_{L\,com} = h_{\mathrm{IV}s} - h_{\mathrm{III}}. \quad (18)$$

Taking into account the internal efficiency of the pump η_{ip} = 0.80 and the mechanical efficiency of the pump η_{mp} = 0.99, the unit work required to drive the circulating pump for the lower cycle l_{Lp} was determined as follows:

$$l_{L\,p} = \frac{l_{L\,com}}{\eta_{ip}\,\eta_{mp}} = \frac{h_{IVs} - h_{III}}{\eta_{ip}\,\eta_{mp}}. \quad (19)$$

The specific isentropic expansion work for the lower cycle was determined as the difference in specific enthalpy before and after the turbine:

$$l_{L\,exp} = h_{\mathrm{I}} - h_{\mathrm{II}s}. \quad (20)$$

Taking into account the internal efficiency of the turbine η_{it} = 0.75 and the mechanical efficiency of the turbine η_{mt} = 0.99, the mechanical unit work of the turbine for the lower cycle l_{Ut} was determined as follows:

$$l_{L\,t} = \eta_{it}\,\eta_{mt}\,l_{L\,exp} = \eta_{it}\,\eta_{mt}(h_{\mathrm{I}} - h_{\mathrm{II}s}). \quad (21)$$

The mechanical power of the lower cycle N_L, taking into account pomp work, was determined as follows:

$$N_L = \dot{m}_L(l_{L\,t} - l_{L\,p}). \quad (22)$$

The electrical power generated in the lower cycle $N_{el\,L}$ was determined by assuming generator efficiency η_g = 0.99 as follows:

$$N_{el\,L} = \eta_g N_L. \quad (23)$$

The heat flux supplied to the lower cycle was determined as follows:

$$\dot{Q}_L = \dot{m}_L(h_{\mathrm{I}} - h_{\mathrm{IV}s}). \quad (24)$$

The electrical efficiency of the lower cycle η_{elU} can be determined from the relationship:

$$\eta_{el\,L} = \frac{N_{el\,L}}{\dot{Q}_L}. \quad (25)$$

3.5. Binary power plant

The electrical power of the binary power plant was determined as the sum of the eclectic powers of the upper and the lower cycles from the following relation:

$$N_{el\,B} = N_{el\,U} + N_{el\,L}. \quad (26)$$

The heat flux delivered to the binary power plant was determined from the following equation:

$$\dot{Q}_B = \dot{m}_w(h_{w1} - h_{w4}). \quad (27)$$

The electrical efficiency of the binary power plant was determined from the following equation:

$$\eta_{el\,B} = \frac{N_{el\,B}}{\dot{Q}_B}. \quad (28)$$

The calculations were simplified by eliminating heat losses to the environment when balancing heat exchangers of the binary power plant. The results obtained for the binary power plant were related to the results obtained for the single-cycle power plant operating in the same temperature range.

4. Results of calculations

The calculations were carried out with the assumption that the mass flux of water supplied from the heat source to the binary or single-cycle power plant is 1 kg/s. This assumption allows the system to be easily recalculated for a different value of the mass flux of water supplied from the heat source. The temperature of the water supplied from the heat source to the binary powerhouse is t_{w1} = 120°C.

The other assumptions used in the calculations are as follows. The superheated steam temperature for the working medium in the upper cycle is t_1 = 110°C (the same temperature value was assumed for the single-cycle power plant), and the condensation temperature for the reference power plant and the lower cycle of the binary power plant is t_{III} = 30°C, and the upper cycle is t_3 = 68°C.

Table 2 shows examples of thermal and caloric parameters at characteristic points of the binary power plant cycle (upper cycle – R413a, lower cycle – R218) and the reference power plant.

Table 3 shows the results of calculations for the electrical power output of the reference (single-cycle) power plant and the binary power plant, which uses R218 refrigerant in the lower cycle. The temperature of the medium at the end of the evaporation process in the upper cycle was t_6 = 93°C, while the superheat temperature in the lower cycle was t_I = 68°C. Also included is the variation of the evaporation temperature of the refrigerant in the lower cycle $t_V = t_{VI}$ from 58°C to 66°C.

Table 2. Thermal and caloric parameters at characteristic points of the circulation of the analyzed power plants.

Single-cycle power plant - zeotropic mixture R413a											
1		2s		3		4s		5		6	
t [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]
110.0	447.9	45.5	418.6	30.0	242.1	31.8	244.5	92.5	356.7	93.0	402.4
Binary upper cycle - zeotropic mixture R413a											
1		2s		3		4s		5		6	
t [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]
110.0	447.9	83.9	416.4	68.0	302.5	70.1	304.1	92.5	356.7	93.0	402.4
Binary lower cycle - zeotropic mixture R218											
I		IIs		III		IVs		V		VI	
t [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]	*t* [°C]	*h* [kJ/kg]
68,0	320.7	36.3	306.5	30.0	233.8	31.3	234.9	66.0	283.4	66.0	316.0
68,0	322.6	38.2	306.5	30.0	233.8	31.2	234.8	65.0	281.5	65.0	316.4
68,0	324.1	39.9	306.5	30.0	233.8	31.2	234.8	64.0	279.8	64.0	316.6
68,0	325.5	41.3	306.5	30.0	233.8	31.2	234.7	63.0	278.1	63.0	316.8
68,0	326.6	42.7	306.5	30.0	233.8	31.1	234.7	62.0	276.5	62.0	316.9
68,0	327.7	43.9	306.5	30.0	233.8	31.1	234.7	61.0	274.9	61.0	317.0
68,0	328.7	45.1	306.5	30.0	233.8	31.0	234.6	60.0	273.3	60.0	317.0
68,0	329.6	46.2	306.5	30.0	233.8	31.0	234.6	59.0	271.8	59.0	316.9
68,0	330.4	47.2	306.5	30.0	233.8	30.9	234.6	58.0	270.3	58.0	316.8

Using the presented calculation methodology and the parameters summarized in Table 2, the electrical powers of the various systems were determined:

- the single-cycle power plant – (A),
- the binary power plant (Fig. 1) – (B),
- the binary power plant with internal heat exchange in the lower cycle (Fig. 2) – (C).

In the case of the reference power plant, the use of internal heat regeneration did not increase the electrical power of the system. When internal heat regeneration was used, the use of energy from the heat source decreased. This means that the return water temperature to the source of the ORC system shown in Fig. 6b is higher than that of the ORC system shown in Fig. 6a.

The calculation results shown in Table 3 indicate that for the binary power plants operating according to the schemes shown in Figs. 1 and 2, the electrical power obtained is lower than for the reference power plant. This is due to the fact that the heat flux to the binary power plant is supplied only in the HE1 heat exchanger. This results in less cooling of the water supplied from the heat source.

In binary power plant systems (B), (C), the heat flux to the lower cycle is supplied in the HE2 condenser-steamer exchanger. Thus, the mass flow of the working medium in the lower cycle $\dot{m}_L$ determined from equation (6) is small, which affects the power of the lower cycle.

The calculation results summarized in Table 3 show that the use of internal heat regeneration in the lower cycle of the binary power plant affects a small increase in the power of the system.

Table 4 shows the values of water temperatures at various points in the cycle for the reference power plant (A) and binary power plants (B), (C). From the data presented in Table 4, it is clear that the temperature of the return water to the heat source in the cases of binary power plants is higher than for the reference power plant. For this reason, the system of the binary power plant (D) operating according to the scheme shown in Fig. 4 was also analyzed. In this system, water from the HE1 exchanger of the upper cycle is directed to the HE4 exchanger of the lower cycle. In the HE4 exchanger, the working fluid of the lower cycle is heated to saturation temperature. This will make the HE2 exchanger use the heat from the upper cycle only to evaporate and superheat the working fluid of the lower cycle. This translates into an increase in the mass flow of the working fluid in the lower cycle (Eq. (8)) and an increase in its power output.

The results of the power calculation of the binary power plant system (D) compared to the single-cycle power plant are shown in Table 5.

Table 3. The electrical power of the single-cycle power plant (A) and the binary power plant (B), (C) for analyzed zeotropic mixture of the upper cycle (t_1 = 110°C, t_6 = 93°C, t_l = 68°C).

t_V [°C]	The electrical power of the power plant [kWe]																	
	R413A			R423A			R426A			R429A			R435A			R437a		
	(A)	(B)	(C)	(A)	(B)	(C)	(A)	(B)	(C)	(A)	(B)	(C)	(A)	(B)	(C)	(A)	(B)	(C)
66.0		10.2	10.6		9.6	10.0		9.6	9.9		8.6	8.9		8.4	8.7		10.2	10.6
65.0		10.1	10.6		9.6	10.1		9.5	10.0		8.6	9.0		8.4	8.8		10.1	10.6
64.0		10.1	10.7		9.5	10.1		9.5	10.0		8.5	9.0		8.3	8.8		10.1	10.7
63.0		10.0	10.7		9.4	10.1		9.4	10.0		8.5	9.0		8.2	8.8		10.0	10.7
62.0	14.8	9.9	10.7	14.1	9.3	10.1	14.0	9.3	10.0	12.1	8.4	9.0	11.7	8.2	8.8	14.9	9.9	10.6
61.0		9.8	10.6		9.2	10.0		9.2	10.0		8.3	9.0		8.1	8.7		9.8	10.6
60.0		9.7	10.5		9.1	10.0		9.1	9.9		8.2	8.9		8.0	8.7		9.6	10.5
59.0		9.5	10.5		9.0	9.9		9.0	9.8		8.1	8.9		7.9	8.6		9.5	10.5
58.0		9.4	10.4		8.9	9.8		8.8	9.7		8.0	8.8		7.8	8.6		9.4	10.4

Table 4. Water temperature at various points in the cycle (working fluid – R413a).

Power plant	Water temperature		
	t_{w1} [°C]	t_{w2} [°C]	t_{w3} [°C]
Reference power plant (A)	120.0	102.5	80.8
Binary power plant (B), (C)	120.0	102.5	92.3

Table 5. The electrical power of the single-cycle power plant (A) and the binary power plant (D) for the analyzed zeotropic mixture of the upper cycle (t_1 = 110°C, t_6 = 93°C, t_l = 68°C).

t_V [°C]	The electrical power [kWe]											
	R413A		R423A		R426A		R429A		R435A		R437a	
	(A)	(D)	(A)	(D)	(A)	(D)	(A)	(D)	(A)	(D)	(A)	(D)
66		17.8		16.8		16.6		14.8		14.3		17.8
65		16.8		15.8		15.6		14.0		13.5		16.8
64		15.9		15.0		14.8		13.3		12.8		15.9
63		15.2		14.3		14.1		12.7		12.2		15.2
62	14.8	14.5	14.1	13.7	14.0	13.5	12.1	12.1	11.7	11.7	14.9	14.5
61		13.9		13.1		13.0		11.6		11.3		13.9
60		13.4		12.6		12.5		11.2		10.9		13.4
59		12.9		12.2		12.0		10.8		10.5		12.9
58		12.4		11.7		11.6		10.4		10.1		12.4

From the calculation results shown in Table 5, it can be seen that for each of the analyzed fluids in the upper cycle in the binary power plant (D), more power was obtained compared to the reference power plant (A).

The highest powers were obtained for the binary power plant (D) with R413a and R437a factors. Analysis of the results for the achieved power for the binary power plant (D) shows that as the evaporation temperature of the refrigerant in the lower cycle increases, the power of the refrigerant increases, causing the power of the entire binary power plant to increase. For evaporation temperatures in the lower cycle lower than about 62°C, the power of the binary power plant is lower than that of the reference power plant.

Analysis of the operation of the binary power plant system (D) showed that it is possible to use the water leaving the HE1 heat exchanger of the upper cycle to carry out the process of heating the medium in the lower cycle to saturation temperature.

This procedure was shown to have a beneficial effect on the power output of the binary power plant.

5. Conclusions

The analysis showed that by properly selecting the parameters of the cycles in a binary power plant, it is possible to obtain more power without changing the heat source than in the case of a single-cycle power plant operating in the same temperature range.

In addition, it was shown that carrying out the process of evaporation of the working medium of the lower cycle in the near-subcritical region causes an increase in the power of the binary power plant.

It follows from the above conclusion that the condensing temperature in the upper cycle should be adjusted to the lower cycle medium in such a way that the condensing temperature in the upper cycle is higher than the critical temperature of the lower cycle medium (this will allow the evaporation temperature in the lower cycle to be adjusted accordingly).

Analysis of the impact of the superheat temperature in the lower cycle showed that an increase in this temperature negatively affects the power output of the binary power plant. For example, for the R413a refrigerant for the superheat temperature in the lower cycle t_{I} = 68°C, the electrical power plant's output was 17,8 kWe, and after the temperature increased to t_{I} =70°C, the output dropped to 17.1 kWe (a further increase in this temperature caused a further decrease in output).

As shown in Table 5, the use of zeotropic mixtures in a binary power plant for each of the analyzed fluids allowed an increase in the electrical power of the power plant compared to a single-circuit power plant. For example, for the binary ORC power plant with R413A refrigerant in the upper circuit, the generated power is 17.8 kWe, which is 20% higher than for a single-circuit power plant (for the reference power plant, the power is 14.8 kWe).

The increase in the power of the binary power plant system was achieved by using the heat contained in the water stream to preheat the medium in the bottom circuit. Such a solution increases the power of the bottom circuit and allows greater cooling of the water, which consequently increases the degree of utilization of energy from the waste heat source.

The established conclusions are consistent with the work analyzed by the authors and are additionally extended to include the use of zeotropic mixture in binary power plants.

In the next articles, the authors plan to conduct an economic analysis of the proposed solution.

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