



Article Thermodynamic Optimization of Low-Temperature Cycles for the Power Industry

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Abstract: The fuel price increase and severe environmental regulations determine energy-saving importance. Useful utilization of low-potential heat sources with 300-400 °C temperature becomes topical. The application of low-temperature power production facilities operating low-boiling heat carriers could be a solution to this problem. A comparative parametric study of a number of heat carriers resulted in a choice of the most promising fluids that are not expensive, have low toxicity and flammability, low ozone depletion and low global warming potential. These heat carriers are considered for application in simple power production cycles with and without regeneration. The main parameters were optimized at the initial temperatures of 323.15-623.15 K. The cycle without regeneration has a maximal net efficiency of 29.34% using the water at an initial temperature of 623.15 K. The regenerative cycle at a temperature below 490 K has its maximal efficiency using a water heat carrier, and at a higher temperature above 490 K with R236ea. The cycle with R236ea at 623.15 K has an electrical net efficiency of 33.30%. Using a water heat carrier, the maximal efficiency can be reached at pressures below 5 MPa for both cycles. Among the organic heat carriers, the minimal optimal initial pressure of a simple cycle is reached with the R236ea heat carrier below 45 MPa without regeneration and below 15 MPa with regeneration. Therefore when utilizing the latent heat with temperatures above 500 K R134a, R236ea and R124 are the most promising organic fluids. Such conditions could be obtained using different industrial sources with water condensation at elevated pressures.

Keywords: organic working fluid; low-temperature; Rankine cycle; Brayton cycle

1. Introduction

The introduction of power-efficient technology is caused by the fuel price increase and the toughening of environmental regulations. The utilization of low-potential heat (LPH) is a promising direction. The application of this type of technology in thermal power plants (TPP) will show remarkable effects.

The TPP thermal cycle parameters and structure determine the LPH source's number and power (Table 1). The gas turbine (GT) exhaust, steam turbine (ST) and Combined Cycle (CC) facilities for flue gas and the back-pressure ST exhaust steam have considerable thermal powers [1,2]. The temperature range of LHP sources typical for the power industry is 50–350 °C [3,4].

Low-temperature power production facilities may be used for useful LPH utilization. Different conversion methods can be used [5], however, the most promising are facilities that operate on different low-boiling fluids including organic heat carriers, carbon dioxide, water–ammonia mixtures, etc. The choice of a low-boiling heat carrier is influenced by numerous factors such as the cycle thermal efficiency, working fluid toxicity, inflammability, etc.



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LPH Туре	Cycle Type	LPH Temperature, $^\circ C$	Hot Stream Components	LPH Share in the Supplied Fuel Heat, %
Gas turbine exhaust	GT	300-500	N ₂ , O ₂ , H ₂ O, CO ₂	20-30
Boiler flue gas	ST, CC	100-200	N ₂ , O ₂ , H ₂ O, CO ₂	6–10
Steam turbine exhaust steam	ST, CC	30–200	H ₂ O	50-60
Coolers (working fluid, oil, coolant)	GT, ST, CC	30-100	Oil, H ₂ O	0.5-1

Table 1. Characteristics of low-potential heat sources in the power industry.

Organic fluids are highly demanded heat carriers for low-temperature power facilities. The power production cycles using organic fluids are called organic Rankine cycles (ORC). The organic working fluids have a high molecular mass which allows for large diameter of the turbine inlet, low rotation speed and erosion-free blades [6]. Besides this, the organic fluids have a high density so the facility dimensions and mass will be low.

The ORC thermodynamic analysis is disclosed in many papers regarding specific use cases. Tchanche et al. [7] considered electric power production from low-temperature solar heat. The working fluids in this paper are RC318, R600a, R114, R600, R601, R113, cyclohexane, R290, R407C, R32, R500, R152a, ammonia, ethanol, methanol, water, R134a, R12, R123 and R141b. The power plant included an intermediate circuit with a heat accumulator and an exit hot flow temperature of 90 °C. At the initial temperature of 348.15 K, the system efficiency varies from 2.61% with R32 to 4.89% with water. Many of the fluids considered in this paper are either toxic or highly flammable. The paper considers only one heat source value that is determined by the heat accumulator design.

Another use of ORC is to use the low-potential heat from different sources presented by the water stream. Sadykov et al. [3] disclosed the utilization of the boiler facility exit heat for the municipal heat supply. The working fluid and the turbine inlet temperature are R245fa and 127 °C, respectively. Additionally, the paper discloses the temperature ranges where different cycles may be preferable. At temperatures below 90 °C, 90–250 °C and 250–300 °C the Kalina cycle, ORC, both organic and steam Rankine cycles, respectively, are more viable and at higher temperatures, only the steam Rankine cycle should be considered.

Galashov et al. [8] used ammonia, butane, pentane, R236fa and R245fa for utilization of the condensate heat downstream of the condensate pump and the latent heat of steam condensation in the steam turbine in the CC in ORC cycles with a regenerator. In the facility with an NK-36ST gas turbine and the heat recovery boiler with 16 MPa and 440 °C inlet steam parameters, the ORC may increase the efficiency by up to 60% at the condensing temperature in the low-potential cycle below 0 °C. The most effective organic fluids are pentane, butane, R245fa, R236fa and ammonia, which are either highly flammable or toxic.

The use of ORC for the utilization of waste heat in the form of gas exhaust was also widely considered in the literature. Douvartzides et al. [9] disclosed the utilization of the exhaust gas heat of a piston engine at 360 °C. The paper considers ethane, R41, propane, R32, ammonia, R152a and other fluids. The maximal and minimal efficiency values of 25.98% and 21.5% are reached with ethane and R152a, respectively. The paper considers specific conditions and does not recommend parameters for different initial temperatures. Many of the fluids considered in this paper are either toxic or highly flammable.

Utilization of the gas turbine exhaust low-potential heat was disclosed by Bălănescu et al. [10]. The Orenda OGT1500 gas turbine has a heat recovery boiler exit temperature of 188 °C, which makes the application of ORC justified. The ORC facility's electric net efficiency of 45.47% and 45.56% was reached using R134a and R123, respectively, which are 1.1% and 1.19% higher than without ORC.

A more general approach was also carried out in the literature. Wang et al. [11] described the use of butane and freons R245fa, R245ca, R236ea, R141b, R123, R114, R113 and R11. The analysis results are optimal cycle parameters for different working fluids. The maximal efficiency of 10.46% is reached using butane with 387.51 K and 1.4 MPa initial parameters. The R245fa working fluid shows a 9.51% efficiency at 380.9 K and

1.4923 MPa initial parameters. The fluids considered in this paper either have remarkable global warming potential or are toxic or explosive. Additionally, in this paper, the initial temperature is below 360 K, which does not include the whole range of temperatures for considered applications.

Li et al. [12] disclosed the choice of an LPH working fluid. The considered fluids were freons R600a, R601a, R600, R601, R245fa, R134a, R236fa and R152a for use in the power cycles with and without a regenerator. The facility using R245fa worked with the 100 °C saturated steam at the turbine inlet. At the heating flow temperature of 120 °C, the efficiency without and with a regenerator was 10.3% and 11.5%, respectively. For R601a, the hot flow and turbine inlet temperatures were 190 °C and 160 °C, respectively, and the efficiency with and without a regenerator was 17.7% and 14.2%, respectively.

Yu et al. [13] considered the operating fluid choice at different parameters of heat to be utilized. It was determined that for a sensible heat source the best was the working fluid with critical temperatures of 25–35 K below the source temperature and, for the sources with temperatures above 393 K among the considered fluids, R601 was the most effective. For the latent heat sources at all temperatures, the best efficiency may be reached using R601 due to its highest critical temperature. Because of this, at all considered temperatures, the turbine inlet steam is saturated. When using a combined heat source, the ratio of latent and sensible heat is an important parameter. At high and low values of this ratio, the working fluid may be taken such as in the boundary case but when the ratio is moderate the working fluid should be chosen for the specific conditions.

Lecompte et al. [14] reviewed the advanced ORC architecture and the heat flow circuits. The analysis results show a shortage of supercritical ORC. The most known are technical, financial and experimental data of subcritical ORC with regenerators, however, the boundary conditions of heat sources vary widely, which makes the analysis difficult.

The ORC are widely used for electricity generation from geothermal, biomass and waste heat energy [15]. The main manufacturer of biomass ORC applications is Turboden, which installed a 6 MW unit in 2015 in Canada using a Rankine cycle with a regenerator [16,17]. The fluid used varies depending on the application. The same Rankine cycle with and without a regenerator is being used by Ormat, with the last project being a 330 MW geothermal power plant in Indonesia in 2018 and the last waste heat recovery project in the USA in 2010 with a 5.5 MW capacity [18].

Thus, we can conclude that most studies on the subject are carried out with specific heat source parameters. Many works use environmental safety as the criteria of the working fluid choice. However, in some cases, fluid toxicity and flammability are not considered, which is not acceptable for the power industry. The objectives of this work are to carry out a justified selection of low-boiling fluids, and optimization of structure and cycle parameters with initial temperatures of 50–350 °C. This temperature range allows for the utilization of exhaust heat in GT and ST boilers, as well as the latent heat of condensation from the back-pressure steam turbine exhaust. At the same time, these initial temperatures include the accepted limits of low-temperature power cycle applications.

2. Low-Boiling-Point Fluid Selection

Published papers consider many refrigerants for the ORC application: RC318, R600a R32, R407C, R290, R500, R134a and R152, R245fa, R245ca, R141b, R11, R123, R113, etc. [3,7,11,19,20]. Depending on various parameters, different organic fluids in similar cycles with initial parameters show different efficiency values. At a maximal cycle temperature of 100 °C the most efficient ones are: R600, R601, R245, R134a, R141b, R11, R123 and R113 [7,11,19].

However, the low-temperature cycles may use simpler heat carriers than organic fluids. The alternative working fluids may be carbon dioxide [21] and ammonia.

The working fluid selection should take into account the following aspects:

- The specific cycle work must be maximal for the taken temperature range;
- The working fluid must have low liquid and vapor phase viscosity to reduce friction losses and increase heat transfer;

- High thermal conductivity will ensure efficient heating and cooling in heat exchangers;
- The working fluid pressure in the cycle must be within a considerable range so as to prevent the difficulties associated with the equipment's strength and tightness;
- Working fluid thermal stability at operating temperatures;
- The working fluid triple point must allow for the avoidance of freezing at operating conditions;
- The working fluid must not be toxic and flammable and any leakage should not hurt the environment;
- The working fluid must be inexpensive and easily accessible.

Thus, it is possible to summarize that the working fluid's important characteristics, besides the thermal efficiency, are the possibility of operation at the given temperatures, operation safety, environmental friendliness and low price. Therefore, in this work, the following fluid selection criteria were taken (Table 2) [7,22–25]:

- Working fluid critical parameters;
- Safety group according to ASHRAE;
- The ozone layer depletion potential (ODP) relating to R11;
- The global warming potential (GWP), i.e., the amount of heat captured in the atmosphere relative to the carbon dioxide capture effect.

The critical parameters of working fluids (Figure 1) are of top importance. At different critical temperatures and equal initial temperatures, a turbine may operate in the area of superheated or humid steam. The heating process may proceed with a constant or growing temperature of the cold heat carrier. Most of the working fluids have critical pressures and temperatures in the ranges of 2–8 MPa and 200–600 K. Water and ammonia have the highest critical pressures of 22.06 MPa and 11.3 MPa. Water also has the highest critical temperature of 647.15 K. The critical parameters' specific features have remarkable influences on the cycle process and efficiency.



Figure 1. Critical parameters of working fluids.

The working fluid application in the power industry causes specific safety requirements. The least dangerous are non-flammable substances, which at the same time have low toxicity. The first condition is caused by the necessity of working fluid operation in turbo-machines and heat exchangers at high temperatures. The compression, heating and friction processes may ignite the working mixture.

Fluid	T _{crit} , K	P _{crit} , MPa	Safety Group	ODP	GWP
R11	471.11	4.41	A1	1.00	4750
R12	385.12	4.14	A1	1.00	10,900
R13	302.00	3.88	A1	1.70	14,400
R14	227.51	3.75	A1	0.00	7390
R21	451.48	5.18	B1	0.04	151
R22	369.30	4.99	A1	0.06	1810
R23	299.29	4.83	A1	0.00	14,800
R32	351.26	5.78	A2	0.00	675
R41	317.28	5.90	A1	0.00	2
R113	487.21	3.39	A1	1.00	6130
R114	418.83	3.26	A1	1.00	10,000
R115	353.10	3.12	A1	0.44	7370
R116	293.03	3.05	A1	0.00	12,200
R123	456.83	3.66	B1	0.02	77
R124	395.43	3.62	A1	0.02	609
R125	339.17	3.62	A1	0.00	3500
R134a	374.21	4.06	A1	0.00	1430
R141b	477.50	4.21	A2	0.12	725
R142b	410.26	4.06	A2	0.12	2310
R143a	345.86	3.76	A2	0.00	4470
R152a	386.41	4.52	A2	0.00	124
R227ea	374.90	2.92	A1	0.00	3220
R236ea	412.44	3.50	A1	0.00	1370
R236fa	398.07	3.20	A1	0.00	9810
R245ca	447.57	3.93	A2	0.00	693
R245fa	427.20	3.64	B1	0.00	1030
R417a	360.19	4.04	A1	0.00	2346
R422a	344.90	3.75	A1	0.00	3143
R422d	352.71	3.90	A1	0.00	2729
R423a	372.64	3.59	A1	0.00	2280
R170 (Ethane)	305.33	4.87	A3	0.00	6
R290 (Propane)	369.83	4.25	A3	0.00	3
R600 (Butane)	425.13	3.80	A3	0.00	4
R600a	408.15	3.65	A3	0.00	4
R601 (Pentane)	469.70	3.37	A3	0.00	4
R601a	460.55	3.39	A3	0.00	0
Cyclohexane	553.64	4.08	A3	0.00	-
Benzene	562.05	4.89	B2	0.00	-
Toluene	591.75	4.13	A3	0.00	-
R717 (Ammonia)	405.40	11.33	B2	0.00	0
RC318	388.35	2.78	A1	0.00	10,300
R718 (H ₂ O)	647.15	22.06	A1	0.00	0
R744 (CO ₂)	304.15	7.38	A1	0.00	1
R729 (Air)	132.83	3.85	A1	0.00	0

Table 2. Low-temperature cycle working fluids' physical and environmental properties.

Low toxicity is important for facility maintenance and repair. Furthermore, a facility operation is always followed by fluid leakages through seals. Significant working fluid emissions may occur during a repair because of a circuit tightness breach. Therefore, the working fluid must have low toxicity in order to avoid creating hazardous working conditions for employees. Otherwise, additional measures must be taken to prevent the component's leakages and the concentrations of individual components in the air, ventilation and to keep the air clean. Table 3 summarizes the toxicity of the ASHRAE safety group A1 fluids chosen for further analysis.

Table 3. Safety groups of working fluids.

		Toxicity		
		A (Low)	B (High)	
Flammability —	1 (none)	R11, R12, R13, R14, R22, R23, R41, R113, R114, R115, R116, R124, R125, R134a, R227ea, R236ea, R236fa, R417a, R422a, R422d, R423a, RC318, R718 (H ₂ O), R744 (CO ₂), R729 (Air)	R21, R123, R245fa	
	2 (low)	R32, R141b, R142b, R143a, R152a, R245ca	Benzene, R717 (Ammonia)	
	3 (high)	R170 (Ethane), R290 (Propane), R600 (Butane), R600a, R601 (Pentane), R601a, Cyclohexane, Toluene	-	

The working fluid leakages mentioned above, together with the 30 years of equipment operation, may be significant depending upon the facility's capacity. This determines the necessity of taking the environmental parameters into account. The smallest potential for ozone layer damage and global warming are the following agents (Figure 2): R21, R22, R32, R41, R123, R124, R134a, R152a, R236ea, R245fa, R245ca, R170, R290, R600, R600a, R601, R717, R718, R744 and R729.



Figure 2. Environmental characteristics of working fluids.

The performance analysis determines the selection of fluids for the LPH utilization in TPP as the following: R22, R41, R124, R134a, R236ea, R718 (H_2O), R744 (CO_2) and R729 (Air) (Table 4, [26–33]). These working fluids have minimal toxicity, flammability and maximal environmental safety.

Table 4. Main parameters of selected working fluids for low temperature cycles.

Fluid	T _{crit} , K	P _{crit} , MPa	T _{max} , K	Safety Group	ODP	GWP	Cost, USD/kg
R22	369.30	4.99	603	A1	0.06	1810	7.97
R41	317.28	5.90	-	A1	0.00	2	12.00
R124	395.43	3.62	573	A1	0.02	609	25.33
R134a	374.21	4.06	641	A1	0.00	1430	8.77
R236ea	412.44	3.50	-	A1	0.00	1370	10.67
R718 (H ₂ O)	647.15	22.06	>2273	A1	0.00	0	0.01
R744 (CO ₂)	304.15	7.38	>2400	A1	0.00	1	0.67
R729 (Air)	132.83	3.85	-	A1	0.00	0	0.00

Figure 3 shows T-S saturation curve diagrams of the selected fluids with the initial temperature range of 323.15–623.15 K, which is typical for the low-potential heat sources in TPP [34]. Most of the fluids have critical temperatures within the selected cycle initial temperatures. Therefore, the Rankine cycle with sub- and supercritical pressures is relevant for these fluids. For the water vapor, all considered cycles were sub-critical and differ only in the presence or absence of superheating. For the air, the only possibility is the use of the Brayton cycle with gas fluid compression in a compressor.



Figure 3. TS diagram of different working fluids.

3. Object of Study

In this work two types of power production cycles were considered:

- The Brayton cycle with air heat carrier (Figure 4);
- The Rankine cycle for all other heat carriers (Figure 5).



Figure 4. Brayton cycle: (a) Scheme; (b) Cycle.



Figure 5. Rankine cycles: (**a**) Scheme; (**b**) Saturated subcritical cycle; (**c**) Superheated subcritical cycle; (**d**) Supercritical cycle.

The power production facility shown in Figure 4 operates in the following way. Air with supercritical parameters expands and produces work from point 1 to point 2 in the turbine. Then, it is cooled from point 2 to point 3 where it gives its heat to the cooling water. The cooled working fluid enters the compressor where it is compressed from point 3 to point 4 and then heated up to the turbine inlet temperature at point 1.

The power production facility shown in Figure 5a can operate at sub- and supercritical working fluid pressures. The Rankine cycles with the turbine operation at sub-critical pressures with saturated and superheated steam, as well as at supercritical pressures, are shown in Figure 5b–d, respectively.

In all cycles shown in Figure 5, the working fluid expands in the turbine (process 1-2), enters the condenser, where it cools down (process 2-2") and condenses (process 2"-3). The resulting condensate is sent to the pump for compression (process 3-4) and then goes to the heat source, where it reaches the initial temperature (process 4-1).

The advantage of the Rankine cycle operating with saturated steam is the possibility of reaching maximal thermal efficiency at the constant heat source temperature and the fixed heat exchanger temperature difference. Its disadvantage is the possibility of low-boiling fluid moisture formation during the expansion. The saturation vapor curve of wet working fluids (R41, CO_2 and others) moves towards the higher entropy as the temperature decreases. When the moisture content becomes remarkable it may considerably reduce the turbine efficiency and life due to erosive wear.

The working fluid superheating above the saturation parameters reduces the risk of moisture formation in the turbine flow path. In the case of constant heat source temperature and heat exchanger temperature difference, the working fluid superheating reduces the mean integral heat supply temperature. Besides this, excessive superheating may lead to the superheated steam turbine exhaust, which increases the mean integral temperature of heat removal. The latter factors may lead to a decrease in the ORC thermal efficiency.

Additionally, this work considers cycles with the regeneration used to reduce losses in the cold source (Figures 6 and 7). In contrast to the simple cycles shown in Figures 4 and 5, here, the turbine exit flow enters the regenerator where it is cooled (process 2-5) by heating the compressor/pump exit flow (process 4-6).



Figure 6. Brayton cycle with regeneration: (a) Scheme; (b) Cycle.



Figure 7. Rankine cycle with regeneration: (a) Scheme; (b) Cycle.

The constant input data used in the analysis are summarized in Table 5 [35,36]. As the main variable parameter, the initial temperature was considered, which varied in the range of 323,15–623,15 K with a 50 K step and the initial pressure. Some of the heat carriers (R22, R124) [37,38] started to decay in the considered temperature range so the initial temperature values were limited for them (Table 4). Considering the Brayton cycle, the final pressure was also varied in the 8–20 MPa range with a 0.5 MPa step.

Table 5. Analysis of input data.

Parameter	Value
Minimum temperature in the cycle, K	303.15
Regenerator pinch point, K	5
Pump/compressor internal isentropic efficiency	0.85
Turbine internal isentropic efficiency	0.85
Power generator and motor efficiency	0.99
Mechanical efficiency	0.99
Minimal stream dryness at the turbine exit	0.86

4. Modeling Methods

Mathematical models of low-temperature power plant cycles were created using the Python 3 programming language. The working fluids thermodynamic parameters were obtained from the CoolProp library, which has calculation algorithms based on solving the Helmholtz free energy equations of state. This method is currently considered to be one of the most accurate [34]. The analysis assumed different turbine inlet temperatures and pressures. The assumptions were that the pressure losses in heat exchangers were zero and the efficiency of the turbine did not depend on the stream dryness fraction.

The main parameters in the cycle were calculated according to [39]. The specific power generated by the turbine was calculated by the equation:

$$l_t = h_1 - h_2 \tag{1}$$

where

 h_1 —turbine inlet enthalpy, kJ/kg;

 h_2 —turbine outlet enthalpy, kJ/kg.

The enthalpy at the turbine exhaust was calculated by the equation:

$$h_2 = h_1 - (h_1 - h_{2t}) \cdot \eta_t \tag{2}$$

where

 h_{2t} —theoretical enthalpy at the turbine outlet kJ/kg;

 η_t —turbine isentropic internal efficiency, %.

The specific power consumed by the compressor or pump was calculated using the equation:

$$l_{c/p} = h_4 - h_3 \tag{3}$$

where

 h_4 —compressor or pump outlet enthalpy, kJ/kg;

 h_3 —condenser outlet enthalpy, kJ/kg.

The enthalpy at the outlet of the feed pump was calculated by the equation:

$$h_4 = h_3 + \frac{(h_{4t} - h_3)}{\eta_{c/p}} \tag{4}$$

where

 h_{4t} —theoretical enthalpy at the feed pump outlet kJ/kg;

 $\eta_{c/p}$ —compressor or pump isentropic internal efficiency, %.

The specific heat supplied to the utilization cycle was calculated using the equation:

$$q_0 = h_1 - h_4$$
 (5)

The net electrical efficiency of low-temperature power plants was calculated using the equation [21]:

$$\eta_{net} = \frac{l_t \cdot \eta_{mech} \cdot \eta_{eg} - \left(l_{s/p} + l_{sp}\right) / \left(\eta_{mech} \cdot \eta_{em}\right)}{q_0},\tag{6}$$

where

l_{sp}—circulation pump work, kJ/kg; η_{mech} —mechanical efficiency, %;

 η_{eg} . —power generator efficiency, %;

 η_{em} —electric motor efficiency, %.

When an input parameters combination produced the turbine exhaust humidity above 14% it was excluded from further analysis [38]. This limitation is assumed in steam cycle analysis but the turbine exit humidity may also occur in the cycles with humid type organic working fluids. Therefore, this criterion was used in the analysis of all Rankine cycles.

To validate the model made for the study the paper, [19] was taken as a basis. The main compared characteristics are presented in Table 6. The input parameters were turbine inlet temperature, condensing temperature and maximum pressure. Because in the mentioned article the BACKONE equation of state was used for determining fluid properties, the minimum pressure in the cycle differs. The same reason can be stated for the differences in turbine outlet temperature and efficiency with and without a regenerator. Overall the calculated parameters are close to the ones mentioned in [19].

There are a lot of different optimization methods which could be applied to power cycles [40]. Among them, one can highlight the simplex method, differential evolution and the dragonfly algorithm, which were already implemented in the literature [41-43]. However, the use of these methods is justified when there are a number of variables. In our case, optimization features only the inlet pressure in the Rankine cycle and inlet and outlet pressure in the Brayton cycle. Therefore, the use of complex optimization methods in our case is excessive. Moreover, the calculation of power cycle parameters in the whole variable range allows for clearer thermodynamic analysis.

Parameter	[19]	Calculations
Fluid	R143a	R143a
Т1, К	373.15	373.15
<i>T</i> ₃ , K	303.15	303.15
p_{\min} , MPa	1.44	1.43
$p_{\rm max}$, MPa	4.5	4.5
<i>T</i> ₂ , K	315.85	312.68
$\eta_{\rm th}$ %	9.05	8.94
$\eta_{ m th}$ with reg., %	9.21	9.10

Table 6. Main parameters of organic Rankine cycles.

5. Results and Discussion

This section discloses the study's results for the air, H_2O , CO_2 , R22, R41, R124, R134a and R236ea working fluids in the initial temperature range of 323.15–623.15K.

Figure 8 shows the analysis results for a simple Brayton cycle with air and inlet temperatures of 623.15 and 473.15 K. The results show that at low initial temperatures the most efficient is the closed Brayton cycle with air. At the initial temperatures of 623.15 K and 473.15 K, the optimal turbine inlet and outlet pressures are 16 and 13 MPa and 16.5 and 15 MPa, respectively. Thus, the optimal initial pressure dependence on the initial temperature is not significant but as the initial temperature decreases the optimal outlet pressure grows. Any deviation in the parameters leads to a decrease in the efficiency of the installation.



(a)

Figure 8. Cont.



(b)

Figure 8. Simple Brayton air cycle efficiency dependence on turbine inlet and outlet pressure: (a) Turbine inlet temperature of 623 K; (b) Turbine inlet temperature of 473 K.

Based on the simulation results of power cycles with H_2O , R124, R134a, R22, R41, R236ea and CO_2 , the net efficiency and dryness at the turbine outlet were determined with various combinations of initial parameters. At the turbine outlet, dryness below 86% of the input parameters combination was excluded from further analysis. For example, the water cycle reaches its maximal efficiency only with expansion into the humid steam area, therefore, it was necessary to remarkably limit the initial pressures (Figure 9). This is due to the high critical parameters combined with the wet-type working fluid [9], which differs from the other heat carriers. For organic coolants, the limitation of the initial pressure did not create significant restrictions.



Figure 9. Turbine outlet dryness fraction dependence on inlet parameters for simple steam cycle.

Figure 10 illustrates the simple Rankine cycle analysis example. It shows the efficiency dependence on the initial temperature and pressure for the R124 heat carrier. At the subcritical initial temperature, maximal efficiency is reached at the turbine inlet pressure equal to the saturation pressure. When the initial temperature is supercritical, the optimal initial pressure is considerably higher and reaches 31 MPa at the maximal allowable temperature of 573 K. This is due to the increase in pump- and turbine-specific power with the increase in initial pressure, which leads to the optimum at the supercritical initial parameters. The dependence of the Rankine cycle efficiency on initial temperature and pressure was calculated for the remaining selected working fluids in a similar way.



Figure 10. Simple Rankine cycle efficiency for R124 and different inlet temperatures and pressures.

Figure 11 shows optimal initial pressures for different working fluids and initial temperatures. Water has the smallest initial pressure at all initial temperatures. The maximal optimal initial pressures are observed when using carbon dioxide and R41. In this range of initial temperatures, other organic heat carriers have similar optimal pressures. This distribution of initial pressures among the Rankine cycle fluids can be explained by the working fluids' critical parameters. Water is used in sub-critical cycles at all initial temperatures because of its high critical temperature. In the case of CO₂ and R41, their critical parameters are similar, which is also reflected in the similarity of the optimal initial pressure. At all initial temperatures, the optimal initial pressure values are in the narrow range of 19–23 MPa. This may be explained by the weak dependence of the air thermodynamic parameters on its pressure because of very low critical parameters. Thus, the minimal cycle optimal pressure may be reached with the working fluid with maximal critical parameters.

Figure 12 shows the dependence of simple cycle efficiency on the initial temperature for different working fluids. As expected, the Brayton air cycle has the smallest efficiency, which is caused by the necessity of gaseous working fluid compression. The Rankine cycle versions have much higher efficiency because of much smaller compression power consumption due to the working fluid liquid state. Among the organic working fluids at low initial temperatures, R236ea and R124 have the highest efficiency. At higher initial temperatures, R22 and R41 have higher efficiency. The water cycle has maximal efficiency because at all initial temperatures it operates at sub-critical parameters which allow the turbine inlet temperature to be near the saturation pressure. Since the minimum steam dryness at the turbine exhaust was previously assumed, in most of the initial temperature ranges, the cycle operates with superheated steam.



Figure 11. Optimal initial pressures for simple cycles at different initial temperatures.



Figure 12. Maximum efficiencies for simple cycles at different initial temperatures.

Most organic fluids have a dry or isentropic shape of the vapor saturation curve. Therefore, their turbine exit steam is superheated and they operate in the sub-critical temperature range with the saturated steam at the turbine inlet. The organic working fluid cycles have smaller efficiency because of the specific features of the heat supply process. After compression in a pump, the working fluid is heated up to the saturation point and then it vaporizes. Accordingly, the heating process in the liquid phase occurs at an increasing temperature that is below the turbine inlet temperature. The evaporation process goes at the turbine inlet pressure, or almost at the same level, and the process temperature may be equal to or a little lower than the turbine inlet temperature.

Thus, due to the water having higher heat of vaporization than organic fluids, the mean integral temperature of heat supply turns out to be higher in the water cycle. In addition, the steam turbine expansion process ends in the humid steam area, therefore, the mean heat removal temperature is lower than in the organic cycles. These two specific features determine the higher efficiency of water cycles at equal initial temperatures.

Below, the analysis results are presented for the considered cycles with regeneration. Figure 13 presents analysis results for the Brayton air cycle with regeneration at the turbine inlet temperatures of 623 and 473 K. Similar calculations were carried out for the whole range of initial temperatures.



Figure 13. Regenerative Brayton air cycle efficiency dependence on turbine inlet and outlet pressure: (a) Turbine inlet temperature of 623 K; (b) Turbine inlet temperature of 473 K.

Figure 14 presents an analysis example of the Rankine cycle with a regenerator using R124. The figure shows the cycle net efficiency dependence on the initial temperature and pressure. Similar calculations were carried out for all Rankine cycle working fluids.



Figure 14. Net efficiency of Rankine cycle with regenerator for R124 and different inlet temperatures and pressures.

Figure 15 shows optimal initial pressures for the cycles with regeneration. In the case of the air cycle, the optimal initial pressure remains high but it decreases with the increase in initial temperature. This is caused by the higher influence of the regenerator at a higher initial temperature. The regeneration share may be increased by the reduction of the initial pressure, which determines lower compressor outlet temperature and higher turbine exit pressure, which increases the available temperature drop.



Figure 15. Optimal initial pressures for cycles with regenerator at different initial temperatures.

Using other working fluids, the optimal initial pressure distribution is similar to the simple cycle one but at high initial temperatures, the values are approximately twice as small. Similar to the Brayton cycle case, the lower initial pressure allows an increase in the regeneration ratio at a higher initial temperature. The optimal initial pressure line has an inflection at 350–400 K associated with the cycle's initial pressure transition from sub- to supercritical. Moreover, this inflection is the least noticeable in the case of substances with low critical pressure.

The regenerative cycle maximal efficiency for different working fluids and initial temperatures is shown in Figure 16. The Brayton air cycle has the lowest efficiency. The Rankine water cycle is the most efficient at the initial temperatures below 490 K. At higher initial temperatures the organic working fluid cycles are more efficient due to the utilization of the regeneration heat.



Figure 16. Maximum efficiencies for cycles with regenerator at different initial temperatures.

Among the organic fluids, the ratio of the efficiencies of cycles is distributed equally in the order of R236ea, R124, R134a, R22, and R41. In the case of water cycles, regeneration is impossible because of the steam expansion at the turbine outlet in the humid steam area. While maintaining the noted trend for higher values of initial temperatures, the efficiency of the carbon dioxide cycle may exceed the efficiency of the steam one.

To calculate the mass flow rate, the heat source power was assumed to be equal to 1 MW. The yearly expense needed to replenish different fluids depending on a leakage in the range of 0.125–2% and 6000 h of operation per year was calculated for the optimal initial pressure and a 473.15 K temperature (Figure 17). The use of air was not considered due to its zero cost. There is a tight group of fluids R41, R134a, R22 and R236ea which have similar yearly expenses due to their close cost per kilogram. The most expensive is the use of R124 because of its high costs. The yearly cost of using carbon dioxide is lower due to its similar mass flow but has a much lower cost compared to the organic fluids. When using water, the mass flow is small due to the high latent heat of vaporization. This, combined with the low-cost fluid, leads to the lowest yearly expense using water.

The calculations above were carried out for various initial cycle temperatures. However, when utilizing low-potential energy sources in the process of heat exchange, the temperature of the hot source will depend on its phase state and may change in the process. Thus, the low-potential cycle initial temperature with different sources having equal maximal temperatures may depend on the properties of the hot source and the heat exchange conditions. Conditions may arise when using a hot source with a variable temperature, a cycle with regeneration will produce less electrical power than a simple cycle, even with a higher net efficiency.



Figure 17. Maximum efficiencies for cycles with regenerator at different initial temperatures.

6. Conclusions

- 1. Water, air, CO₂, R22, R41, R134a, R124 and R236ea are the most suitable working fluids for the utilization of low-temperature heat in the energy sector in terms of environmental safety, flammability and toxicity.
- 2. Cycles using water vapor have the lowest optimum initial pressure, followed by R236ea, R124, R134a, R22, R41 and CO₂. The maximal initial optimum pressure of 63 MPa in the simple cycle is reached using carbon dioxide, in the regenerative cycle, this pressure is 28.5 MPa. In the simple and regenerative Brayton air cycles, the optimal pressures are 20–25 MPa and 15–20 MPa, respectively.
- 3. The maximum efficiency in a simple cycle is achieved over the entire range of initial temperatures when using water. Its maximal efficiency of 29.34% is reached at 623.15 K. Among organic working fluids, R236ea, R124 and R134a are the most efficient at low initial temperatures below 450 K, and R22 with R41 at higher initial temperatures above 520 K. Carbon dioxide shows higher efficiency than R236ea at the initial temperature of 623.15 K. The simple air cycle has 1.01–6.65% efficiency at 473.15–623.15 K inlet temperature, at lower temperatures the efficiency is less than zero.
- 4. When used in a regenerative cycle and with starting temperatures of up to 490 K, the plant is most efficient when using water. At a higher initial temperature above 490 K, higher efficiency is observed with R236ea, which reaches 33.30% at a 623.15 K temperature. The regenerative air cycle at 423.15 K–623.15 K has 0.11–21.87% efficiency. At lower temperatures, the efficiency is less than zero.

In future studies, we are planning to consider the heat source-specific characteristics during the thermodynamic analysis and optimization to improve the applicability.

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