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Influence of the temperature difference between the heat source and the evaporation temperature in ORC systems working with natural refrigerants

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Abstract

The use of waste heat in many branches of industry is limited due to temperature in the range of 30 to 100^{o}C. One of the methods of using waste heat are devices that implement the Organic Rankine Cycle (ORC). In currently used ORC systems, the heat source temperature is at least 80^{o}C, while the low temperature heat source (usually atmospheric air) has a temperature of 30^{o}C. The work analyzes the influence of the organic fluids properties on the performance of the proposed installation driven by the waste heat and working based on the ORC. The basic operation parameters in nominal conditions were determined for three selected natural refrigerants R290, R600a, R717 and one synthetic R245fa. The condensing temperature 30^{o}C were defined as a nominal value. The research results compare how the generated electric power will change depending on the temperature difference between the temperature of the heat source and the temperature of evaporation. It turns out that for a device with finite dimensions, the maximum power is obtained for a specific evaporation temperature. And this is not the highest temperature that can be achieved. The highest evaporation temperature allows for the highest efficiency of the system, but not the maximum of capacity.

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Keywords: Organic Rankine Cycle; refrigerant; waste heat; working fluid

1 Introduction

The ORC is an unconventional and very promising technology for electricity generation that has been gaining increasing popularity [3, 4, 13, 15]. ORC-based systems employ the steam Rankine Cycle except that instead of water, organic or inorganic fluids characterized by a low boiling point are used as the working fluid [14, 15].

Despite numerous advantages, including high latent and specific heat, chemical stability in a wide range of temperatures, low viscosity, non-toxicity, non-flammability and availability, water has a peculiar disadvantage, namely a high normal boiling point of 100^{o}C, which renders water unusable in practice in low-temperature systems [22, 25, 28, 31] . On the other hand, organic refrigerants are characterized by a low normal boiling point, often below 0^{o}C [7].

There is a lot of research studies on selecting an optimum fluid for use in certain conditions [1, 24, 26, 30, 35]. In Ref. [12], the effect of working fluid properties on the irreversibility of a cycle supplied by a 10 MW waste heat source at 600 K is analyzed. Among the examined refrigerants the R113 and R123 were characterized by the highest cycle irreversibility which, at the same time, decreased at a lower rate with increasing evaporation pressure compared to the other fluids. Studies presented in Ref. [16] focus on the effect of the evaporation pressure of six chosen refrigerants on the power output of a Rankine cycle supplied from a heat source with temperature varying from 80^{o}C to 200^{o}C and the condensation temperature of 20^{o}C. It was found that an increase in pressure leads to an increase in the system power output to the specified maximum value for all the fluids. The highest power output was achieved for R227ea between 80^{o}C and 160^{o}C, and for R245fa between 160^{o}C and 200^{o}C. In Ref. [22], Mikielewicz recommends some working fluids for use in residential ORC systems for generating 4 kW of heat and electricity. Among proposed fluids at the turbine inlet temperature of 170^{o}C and the condensation temperature of 50^{o}C, R123, R141b and ethanol achieved the

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highest efficiency. In the research presented in Ref. [34], heat transfer area minimization and heat recovery efficiency served as criteria for comparing the following fluids: R123, R134a, R141b, R142b, R152a, R227ea, R236ea, R236fa, R245ca, R245fa, R600, R600a and n-pentane [20, 33]. Also in this case the results were related to the heat source temperature. R123 performed as the best fluid between 100^{o}C and 180^{o}C, and R141b for heat sources at over 180^{o}C. In Ref. [21], an attempt was made to select a working fluid which for a given supply temperature, varying in the study from 150^{o}C to 250^{o}C and at constant condenser cooling water temperature of 5^{o}C will provide maximum power output. Some of fluids operated in a supercritical cycle. The highest power output between 150^{o}C and 175^{o}C was achieved for R134a, between 175^{o}C and 185^{o}C for R142b, between 185^{o}C and 210^{o}C for R245fa, between 210^{o}C and 225^{o}C for R245ca, between 225^{o}C and 230^{o}C for R365mfc, and between 230^{o}C and 250^{o}C for R123. In turn, Schilling et al. concludes that the choice of the refrigerant should be directly connected to the ORC installation design [29]. In this case, interventions resulting from subsequent design will be avoided. Many studies have examined the influence of the properties of the single ORC component working fluids as well. Table 1 summarizes refrigerant recommendations depending on the temperature of the heat source.

In addition to single-component working fluids, mixtures can also be used in ORCs. Such fluids make it possible to match the fluid to the heat source more effectively and to develop a substance exhibiting desirable properties, taking into account environmental and safety requirements. The temperature of heat sources that supply systems often varies. A characteristic of single-component substances is a constant phase change temperature. This is not the case with multi-component, zeotropic fluids, which in the process of phase change are subject to a so-called temperature glide [5, 27, 29]. As the fluid temperature changes when heat is supplied, the temperature profile of the refrigerant can be better aligned to that of the heat source [18]. In this way, heat and exergy losses are lower, while the system efficiency is higher [19]. Such positive consequences are possible only for a certain composition of the zeotropic mixture and thus for a particular temperature glide [36].

Table 1. Recommended refrigerants according to heat source temperature

Source temperature [^{o}C]	Recommended refrigerant
<100	R32, R125, R134a, R143a,
100-120	R124, R227ea, R290, R1234yf, R1270
120-160	R114, R141b, R123, R124, R245fa, R601a, R1243ze
160-200	R123, R141b, R1234ze, RC218, R236fa, R236ea, R600, R601
>200	benzene, paraxylene, toulene, hexane

Zeotropic refrigerants were also investigated in a number of studies. Liu et al. [11] analyze the effect of the composition of R600a/R601a and R227ea/R245fa mixtures on the efficiency of an ORC supplied by variable-temperature geothermal energy, with cooling water at 20^{o}C as the lower heat source. For the molar ratio of 90% of R600a in an R600a/R601a mixture, an increase of 8% in efficiency was achieved compared to the pure fluids. The maximum efficiency with an R227ea/R245fa mixture occurred for the molar ratio of 80% of R227ea in the mixture, and was higher by 15% compared to a system employing a single-component refrigerant. The increase in efficiency compared to single-component refrigerants was achieved only for temperatures below 120^{o}C; above that value, R227ea and R600a provided higher efficiency. In Ref. [5], an ORC system with pentane/hexane and R245fa/R365mfc mixtures was studied. The increase in efficiency for high source temperature varying from 150^{o}C to 250^{o}C was from 6% to 16%, while electric power increased by 20%. It was also demonstrated that using three-component mixtures in a system has no significant impact on improving the efficiency. In Ref. [19], the effect of the temperature glide of R600/R601, R600/R601a, R600a/R601, and R600a/R601a mixtures on the operation of an ORC system was studied. The results has shown that the increase in the efficiency of a system using mixtures compared to a system employing single-component refrigerants is possible only for a temperature glide that is lower than a change in the temperature of the fluid supplied to the evaporator. Similar conclusions were presented in Ref. [36], which demonstrated that the temperature of the heat source determines the selection of the mixture. As a results from the research is fact that for a constant temperature range the efficiency and output power of a system employing single-component refrigerants are higher than those of a system working with mixtures. Another approach for choosing the correct mixture as a working medium was proposed by Oyewunmi and Marides in the publication [23]who as a criterion applied the cost of the construction of the installation and its subsequent operation. In their studies, it was shown that the installation driven by heat source of temperature 98^{o}C from the thermodynamic point of view should be used mixtures of 50%

n-pentane and 50% n-hexane or 60% R-245fa and 40% R-227ea. However, for clean components, the installation cost is 14% cheaper. The other results were obtained by Le, Kheiri et al. [17], whose economic analysis has shown that the use of pure refrigerant R245fa is the least viable. The theoretical analysis assumes that the system generates 1 kW of electric power. And on the basis of this input data, the other elements of the system were selected. Unamba C.K. et al. for a typical ORC installation with an electric output power of 1 kW, used the R245fa as refrigerant obtained a thermal efficiency of the system from 0 to 6% depending on the working conditions [32].

2 Description of the proposed installation

The paper aims to select an optimal working fluid and optimal temperature difference for the proposed installation schematically depicted in Fig. 1. The working fluid in the cycle of the system under consideration is subject to the transformations: isobaric evaporation, expansion, isobaric condensation, and liquid compression [11, 32].

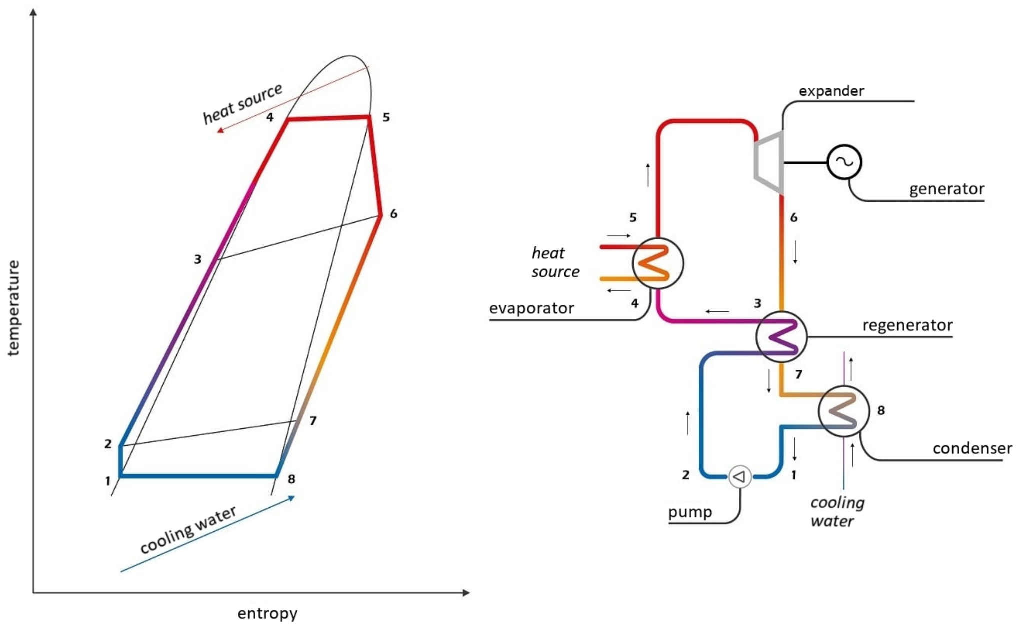

Figure 1. Schematic illustration of the proposed installation

3 Modelling approach

Basic thermodynamic transformations for the proposed ORC system are presented in Fig. 2. In order to determine the basic parameters of the experimental system, it was additionally assumed that system is in a steady state. Pressure drops in the evaporator, condenser and other elements of the installation are negligible, also the heat losses to the environment are negligible.

The efficiency of all items including turbines and pumps are known:

- isentropic turbine efficiency: $\eta_{TS} = 0.92$;
- mechanical efficiency of the turbine gear: $\eta_{TM} = 0.95$;
- electrical efficiency of the generator: $\eta_{GEL} = 0.93$;
- power output system efficiency: $\eta_{TW} = 0.98$;
- isentropic pump efficiency: $\eta_{PS} = 0.85$;
- pump gear efficiency: $\eta_{PM} = 0.95$;
- electrical efficiency of the pump engine: $\eta_{PEL} = 0.98$.

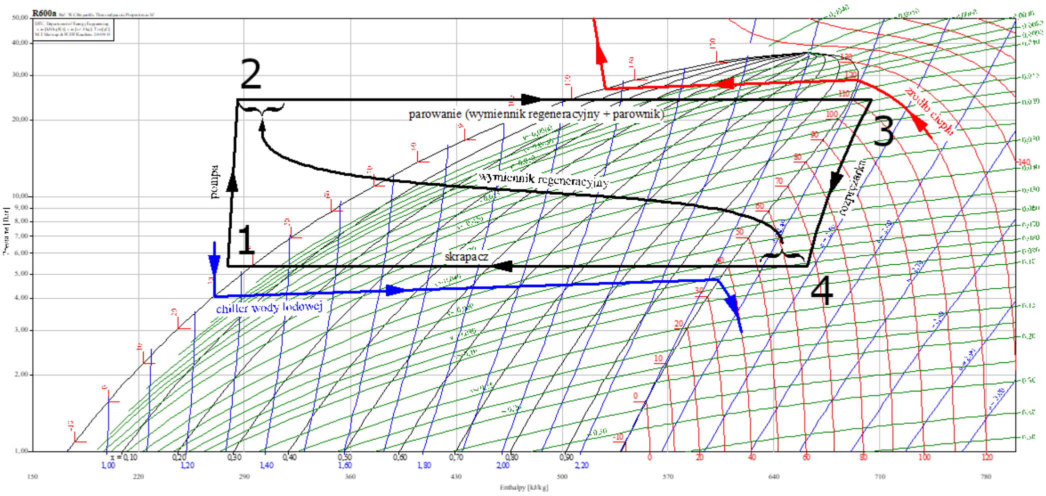

Figure 2. Pressure-enthalpy diagram for a theoretical cycle with dry refrigerant.

Operation cycle starts from position 1 (Fig. 2.). The power N_p required for increasing the pressure from condensation p_C to evaporation p_E level is equal to:

$$N_P = \frac{\dot{m}_R \left(h_{2(s=const)}\right) - h_1}{\eta_P} \tag{1}$$

where pump efficiency:

$$\eta_P = \eta_{PS} \cdot \eta_{PM} \cdot \eta_{PEL} \tag{2}$$

while the enthalpy at point 2, at the end of the pumping process, is equal to:

$$h_2 = h_1 + \frac{\left(h_{2(s=const)} - h_1\right)}{\eta_{PS}} \tag{3}$$

Next step in cycle is evaporation process – line 2-3 in Fig. 2. The heat flux supplied in this process must firstly warms up the refrigerant to the phase change temperature and then evaporates it. Required heat flux is described as:

$$\dot{Q}_{HS} = \dot{Q}_{HS,h} + \dot{Q}_{HS,e} = \dot{m}_R \left(h_3 - h_2\right) \tag{4}$$

where is $\dot{Q}_{HS,h}$ heat flux needed from warimg up refrigerant and $\dot{Q}_{HS,e}$ is heat flux needed for evaporating:

$$\dot{Q}_{HS,h} = \dot{m}_R \left(h_{2'} - h_2\right) \tag{5}$$

$$\dot{Q}_{HS,h} = \dot{m}_R \left(h_3 - h_{2'}\right) \tag{6}$$

For the classic ORC system, the next step in cycle is the expansion in the turbine, in which the work is equal to:

$$\dot{W}_{ORC} = \eta_{TS} \dot{m}_R \left(h_3 - h_{4s}\right) \tag{7}$$

the electric power at the terminals of the generator is:

$$N_{EL} = \eta_{TM} \eta_{TEL} \eta_{TWM} \dot{W}_{ORC} \tag{8}$$

and total efficiency of the ORC system is equal to:

$$\eta_{ORC} = \frac{N_{EL} - N_P}{\dot{Q_{HS}}} \tag{9}$$

In the solutions, in the proposed installation of ORC system, the expanded refrigerant flows into the condenser, where heat is transferred out of the system:

$$\dot{Q_C} = \dot{m}_R \left(h_4 - h_1\right) \tag{10}$$

The end of the expansion of "dry" fluid (difference between dry, wet and isentropic fluids are shown in Fig. 3.), shown in Figure 2, is located in the area of superheated gas. Heat transferred in condenser outside of system can be divided into two heat fluxes: $\dot{Q}_{C,ch}$ - chilling the gas to the condensation temperature and $\dot{Q}_{C,cnd}$ - real condensation process:

$$\dot{Q}_{C,ch} = \dot{m}_R \left(h_4 - h_{4'}\right) \tag{11}$$

$$\dot{Q}_{C,cnd} = \dot{m}_R \left(h_{4'} - h_1\right) \tag{12}$$

In the case of "wet" fluids operating in circuits without superheated gas, the end of the expansion is located in the area of wet gas. The heat transferred to environment is therefore equal to the heat of condensing the refrigerant from a given degree of dryness to a saturated liquid. For "isentropic" fluids, this heat is equal to the total heat of condensation for a given temperature.

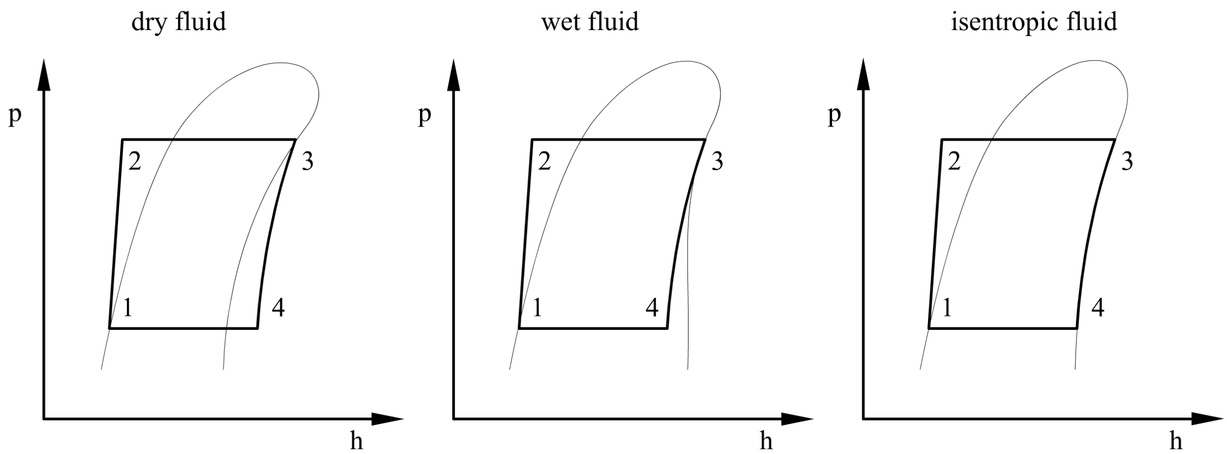

Figure 3. ORC diagram for dry, wet and isentropic fluids

4 Results and analysis

In the first step of the calculations, it was decided to determine how the system operating parameters will differ for the same operating conditions.

- evaporation temperature: 80^{o}C;
- condensation temperature: 30^{o}C.

The selection of the refrigerant for a given temperature range is sometimes based on the vapour saturation curve in the T-s diagram [2, 4, 6, 8–10]. In the group of refrigerants considered, in nominal operating conditions of the set-up, R290, and R717 are wet fluids, while R245fa, and R600a are dry fluids (Tab. 2).

In the group of fluids under consideration, all the wet refrigerants are characterized by higher working pressures than the dry ones. Ammonia, propane (Tab. 2) are characterized by the highest evaporation and condensation pressures in

Table 2. Basic cycle parameters

Quantity	R290	R600a	R717	R245fa
ODP	0	0	0	0
GWP	3	4	0	1050
Safety group	A3	A3	B2	B1
Type of refrigerant	Wet	Dry	Wet	Dry
Evaporation pressure p_p, kPa	3139.9	1343.8	4142	789.3
Condensation pressure p_k, kPa	1079	404.7	1167.2	177.8
Heat of evaporation q_p, kJ/kg	189.8	252.58	873.96	152.51
Heat of condensation q_p, kJ/kg	326.71	323.33	1144.41	187.33
Average specific heat of liquid c_w for p_p, kJ/(kg·K)	3.12	2.66	4.78	1.40
Average specific heat of vapour c_p for p_s, kJ/(kg·K)	3.19	1.87	2.05	0.95
Specific volume of saturated vapour v_3, m^3/kg	0.0118	0.028	0.0295	0.0227
Vapour dryness fraction at the end of expansion	0.948	1.00	0.87	1.00
Refrigerant post-compression temperature T_2, ^{o}C	31.9	30.6	31.0	30.3
Refrigerant post-expansion temperature T_4, ^{o}C	30	40.3	30	40.5
Volumetric flow rate of refrigerant V_3, l/s	0.61	1.34	0.42	2.05
Mass flow rate of refrigerant m_R, kg/s	0.0515	0.0479	0.0142	0.0902
Heat for heating up liquid Q_{dp}, kW	8.00	6.30	3.59	6.28
Heat of evaporation Q_p, kW	9.76	12.08	12.44	13.76
Heat source capacity Q_{GZC}, kW	17.76	18.38	16.03	20.04
Heat sink capacity Q_{DZC}, kW	15.94	16.39	14.17	17.83
Refrigerant pump capacity P_{PC}, kW	0.275	0.104	0.090	0.053
Efficiency of the proposed installation η_D , %	8.57	9.27	9.93	9.50

nominal operating conditions. On the other hand, R245fa and R600a are refrigerants with the lowest evaporation and condensation pressures. Although R245fa features the lowest values of working pressures and pressures ratio is the highest; for R290 the pressures ratio is the lowest. It should be noted that in none of the cases the condensation pressure is lower than the atmospheric pressure, which will protect the system from the possibility of moisture condensation due to leakages.

Ammonia has the highest heat of evaporation and condensation (Tab. 2); these values are almost three times higher than for the other fluids. The lowest heat of phase change in the temperature range considered is observed for R245fa. One quantity that is essential for the proposed installation is the specific volume of vapours of the refrigerant flowing into the flow tanks. It is the value that determines the volumetric flow rate of the glycerol flowing through the expansion device. Among the organic substances considered, ammonia R717 features the highest specific volume of saturated vapour under the evaporation pressure. Slightly lower values of over 20 dm^3/kg are also observed for R600a and R245fa. The volumetric flow rate can also be expressed as a ratio of the system power output (1 kWe in the considered case) to the difference between evaporation and condensation pressures. The larger the pressure difference for a constant power output of the system, the lower the volumetric flow rate of the intermediate liquid is required. In the group of considered fluids, a system employing ammonia requires the lowest flow rate of the liquid, since the pressure difference for ammonia is the largest. As with working pressures, R290 (Tab. 2) are next and require a higher flow rate of slightly over 0.5 l/s. On the other hand, if refrigerants with the smallest difference between evaporation and condensation pressures (R245fa and R600a) are used in a system, the highest volumetric flow rate is required (Tab. 2). According to the pressure values, a relation can be observed such that the volumetric flow rate of the wet refrigerants is lower than that of the dry ones.

The efficiency of proposed ORC system is equal to the ratio of the electric power output minus the power needed for drive the refrigerant pump to the heat supplied. The system efficiency increases when the pump power and the amount of heat supplied to the system are smaller. Among the selected organic fluids, the lower capacity of the fluid pump is required for R245fa (the smallest pressure difference and a relatively low volumetric flow of the liquid refrigerant through the pump) and R717 (the lowest rate of volumetric flow of the liquid through the pump and the largest pressure difference). Such a low value of the rate of volumetric flow of the fluid through the pump and the small pressure difference in the case of R245fa are crucial for the pump driving power. The highest power of the refrigerant pump is required in a system using R290, i.e. refrigerant with a high volumetric flow rate of the liquid. These are also fluids with the highest temperature at the end of the pumping process (Tab. 2). The high value of energy needed to drive the pump may result from the fact that it is partly used for heating the liquid. The least amount of heat drawn from the heat source is a characteristic of ammonia, which has the highest heat of evaporation compared to the other organic fluids. The small amount of heat is a consequence of a very low mass flow rate of R717. It is followed by R600a and R290 i.e. refrigerants with a relatively high heat of phase change.

Due to a low required power for the liquid refrigerant pump and a small amount of heat provided from the heat source, among all the considered fluids it is ammonia that guarantees the highest efficiency of the proposed ORC system (Fig. 4). It is followed by isobutane in terms of efficiency of the proposed installation Despite a relatively small pressure difference and due to a high specific volume of saturated vapour and thus a small mass flow rate and a small amount of heat drawn, R600a provides the efficiency of 9.27%. In the case of the conventional ORC system, second best efficiency, equal to 9.50%, is achieved for R245fa. This is caused by a lower power of the refrigerant pump and the same amount of heat drawn from the heat source compared to isobutane. R290 provides a slightly lower efficiency than that of the proposed installation using isobutane.

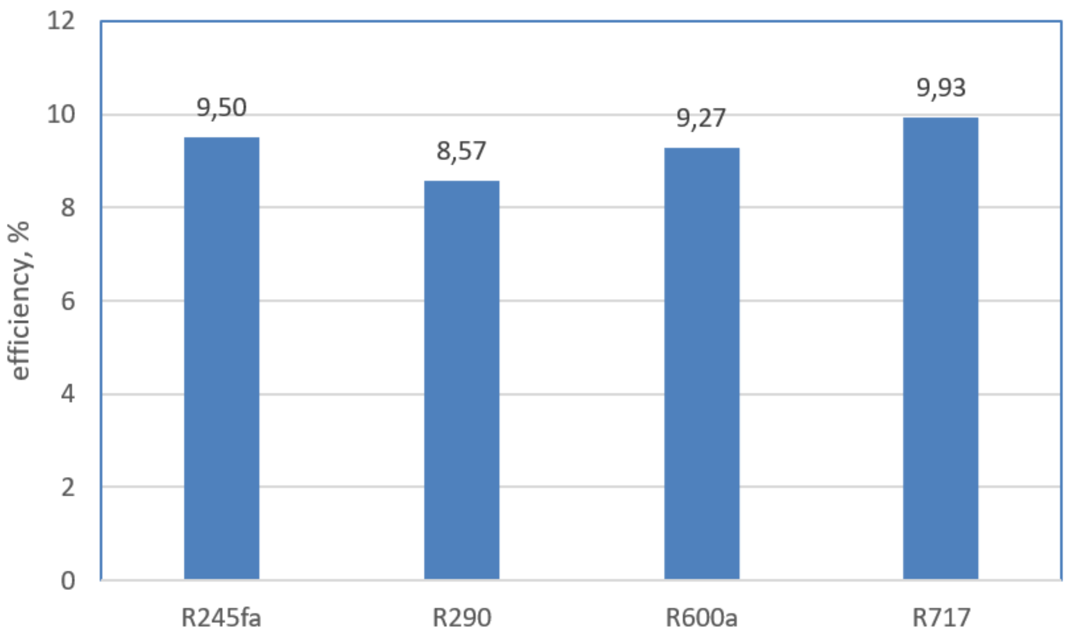

Figure 4. Theoretical efficiency of the proposed ORC installation

4.1 Determining the optimal temperature difference

The next step of this research was to determine the optimal boiling point of the refrigerant for the known temperatures of the heat source and heat sink.

The following constant values were adopted in the calculations: temperature of the heat source, temperature of the heat sinke, mass flux of the heat source, mass flux of the cooling water. For such values, an analysis was carried out on how the efficiency and the generated electric power at the output from the ORC system will change depending on the maintained refrigerant evaporation temperature.

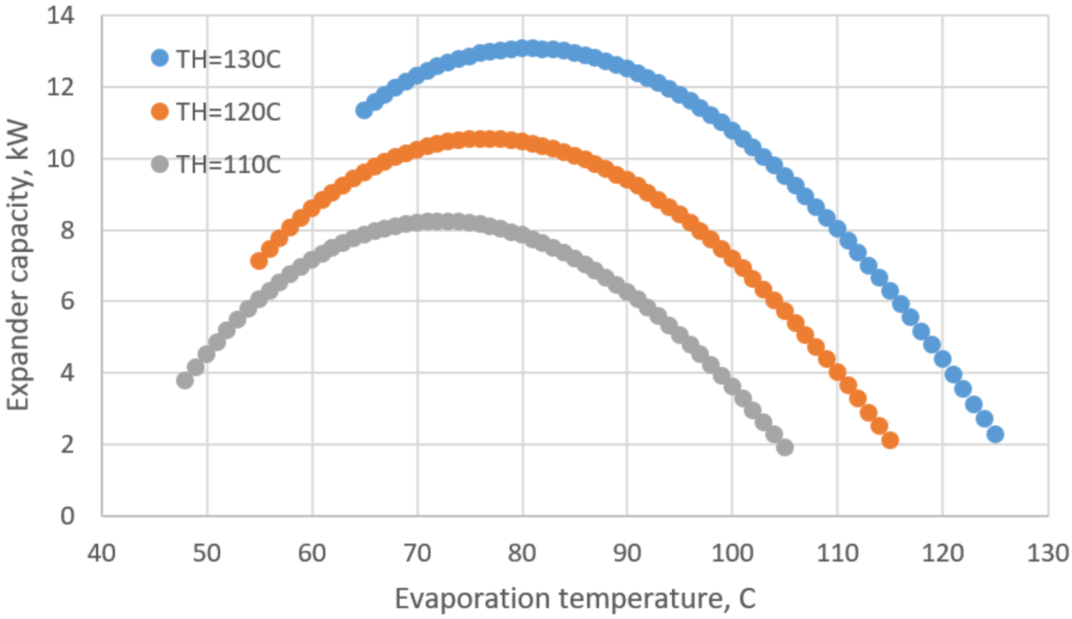

Figure 5. ORC electric capacity for R600a as a working fluid

Figure 5 shows the results for the working medium R600a (isobutane). As it is shown in the diagram, when the heat source is equal to 130°C, the optimal evaporation temperature should be equal to 81°C. For heat source temperature equal to to 110°C, the optimal evaporation temperature should be equal to 72°C.

In Figure 6 there are shown results for R290 (propane) as a working fluid. In this case the optimal values of evaporation temperature are obtained too. For heat source temperature equal to 130°C, the optimal evaporation temperature should be equal to 78°C. For heat source temperature equal to 110°C, the optimal evaporation temperature should be equal to 71°C. Results are limited to 95°C becuse propane critical temperature is equal to 96.7°C. ORC sysems can work in supercritical cycles, but in this research it was decided to compare just undercritical cycles.

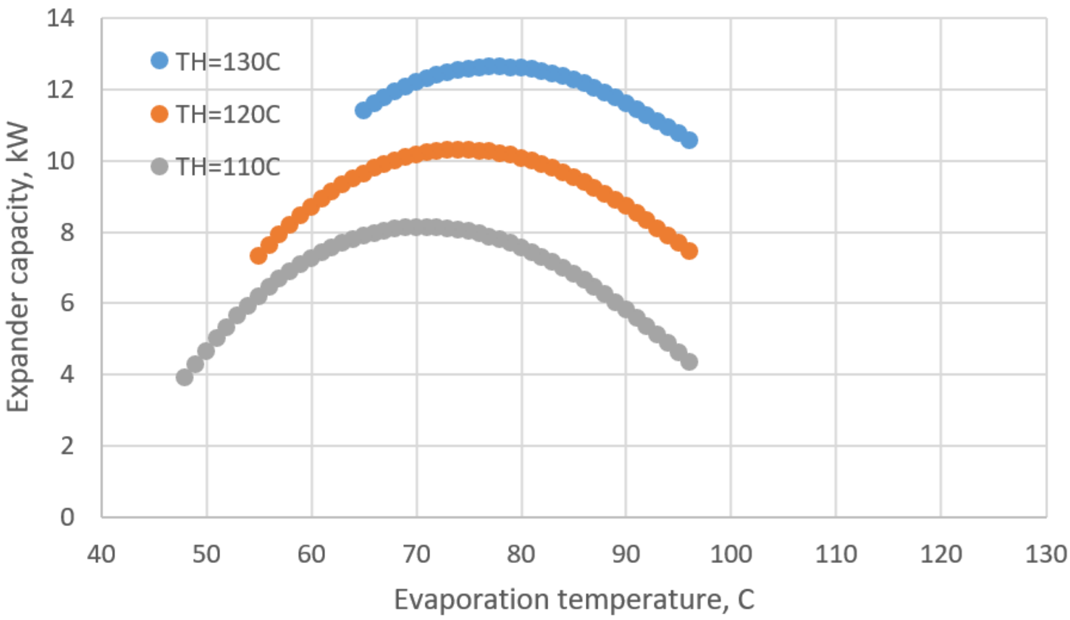

Figure 6. ORC electric capacity for R290 as a working fluid

In Figure 7 there are shown results for R717 (ammonia) as a working fluid. In this case also optimal values of evaporation temperature are obtained. For heat source temperature equal to 130°C, the optimal evaporation temperature should be equal to 80°C. For heat source temperature equal to 110°C, the optimal evaporation temperature should be equal to 72°C. And finnaly in Figure 8 there are shown results for R245fa as a working fluid. In this case also optimal values of evaporation temperature are obtained. For heat source temperature equal to 130°C, the optimal evaporation temperature should be equal to 81°C. For heat source temperature equal to 110°C, the optimal evaporation temperature should be equal to 72°C.

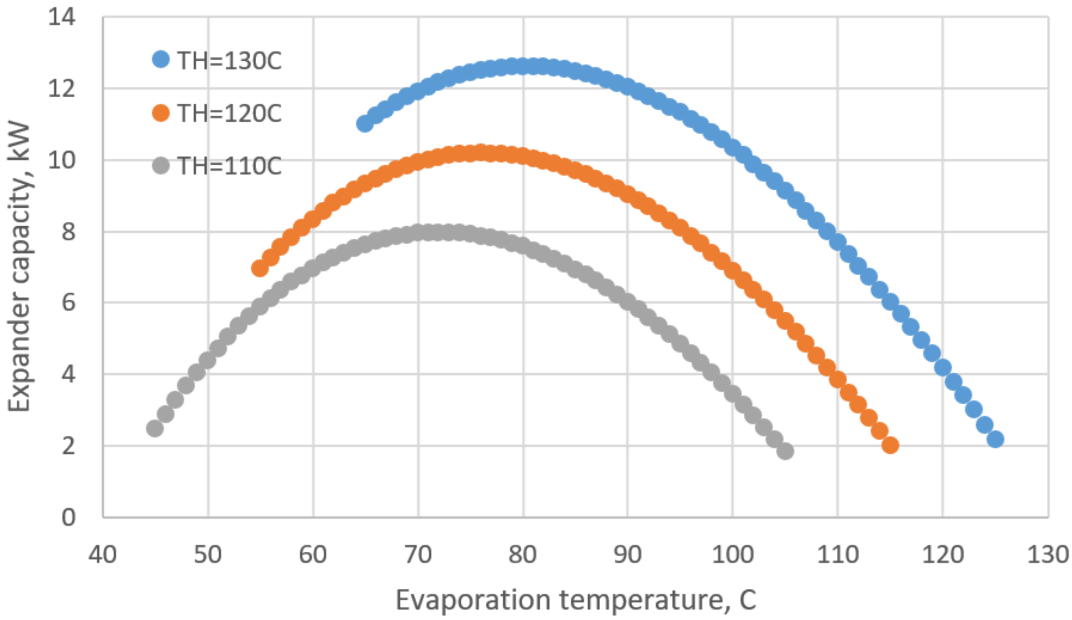

Figure 7. ORC electric capacity for R717 as a working fluid

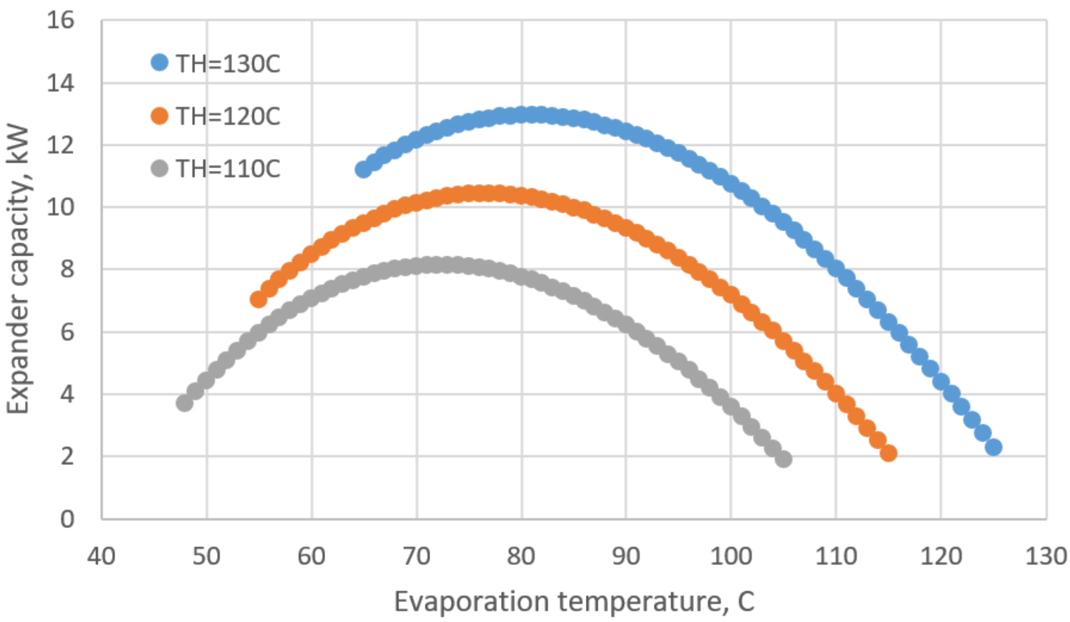

Figure 8. ORC electric capacity for 245fa as a working fluid

Table 3 presents the optimal working medium evaporation temperature for selected in this reserch refrigerants.

Table 3. Optimal evaporation temperature for selected refrigerants

Heat source temperature	R290	R600a	R717	R245fa
130°C	78.0°C ΔT=52K	81.0°C ΔT=49K	80.0°C ΔT=50K	81.0°C ΔT=50K
120°C	74.0°C ΔT=46K	76.5°C ΔT=43.5K	76.0°C ΔT=44K	77.0°C ΔT=43K
110°C	71.0°C ΔT=39K	72.0°C ΔT=38K	72.0°C ΔT=38K	72.0°C ΔT=38K

5 Conclusion

The working fluids in ORC systems are subject to the same regulations as the refrigerants. It is mainly about restrictions related to the ozone depletion potential and the potential to create a greenhouse effect. For this reason, this study focuses on natural fluids. These fluids are safe for the environment, but most often they are dangerous to humans. They are either flammable / explosive or toxic. By applying basic safety rules during design and construction, devices with hazardous working media can also work safely.

In real ORC systems, the investor has a finite heat source and a finite heat sink. And for such input data the ORC system should be selected. The results of the research carried out clearly show that the correct design of the system is not based on the achievement of the highest efficiency, but on the achievement of the highest power generated on the turbine rotor. It turns out that for each working medium, it is profitable to reduce the efficiency by reducing the evaporating temperature in the heat exchanger cooperating with the heat source in order to obtain more power.

For all tested working fluids, the optimal operating parameters of the ORC system were obtained for known heat source temperatures and heat source mass streams. For the natural refrigerants analyzed in the research: R290, R600a, R717, it turned out that when the heat source temperature is 130°C, the evaporation temperature of the working fluid temperature should be around 80°C. And in the case when the heat source temperature drops up to 110°C, the optimal evaporation temperature is 70°C.

Acknowledgements

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