THE METHOD OF THE WORKING FLUID SELECTION FOR ORGANIC RANKINE CYCLE (ORC) SYSTEM WITH VOLUMETRIC EXPANDER

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ABSTRACT

Volumetric expanders are nowadays used in micro, small and medium power ORC systems. Tchanche (2011) indicated that most often spiral, screw and the rotary vane machines are applied. Volumetric machines have a number of specific features determining their operation. The most important are: the possibility of building expanders for small and very small capacities; small and moderate frequency of operating cycle - allowing for consideration of the processes taking place in the machine as a quasi-static; the ability to operate at high pressure drops in a single stage and ease of the hermetic sealing. The most important feature of the volumetric expander operation is the relationship of the expander power and the expansion ratio (the ratio of the inlet and outlet pressure). Each type of volumetric expander also has the optimum value of the expansion ratio. Unlike the turbines, volumetric expanders can operate at low working fluid flow rates and lower pressures. Thus, it is possible to apply volumetric expanders in ORCs powered by low-temperature heat sources, such as e.g. domestic waste heat. The task of a suitable working fluid selection to the ORC system with volumetric expander should be considered differently than in the case of the turbine-based systems. It is caused by low thermal parameters of the cycle and indicated earlier volumetric expander characteristic features. In this paper a new method of the working fluid selection to the ORC system working with volumetric expander was presented. The method is based on the dimensionless parameters useful for the comparative analysis of different working fluids. Dimensionless parameters were defined for selected thermal properties of the working fluids, namely the ability of heat absorption from the heat source, heat removal, mean temperature of the heat supply and the efficiency of the energy conversion. These comparative parameters were calculated for selected low-boiling ORC working fluids and selected temperature of the heat source and the heat sink. Basing on the values of these parameters the working fluids comparison was presented and applicable working fluids were selected.

1. INTRODUCTION

As it was indicated by Bao and Zhao (2013) the most important problems connected with ORC system design are the suitable working fluid and expander selection. They also showed that currently there is a wide range of applicable working fluids available. Expander selection is mainly based on the system power and its purpose. In general two types of expanders can be applied in ORC systems. One are the turbines, the others are volumetric expanders.

Turbines are mainly applied in an large power (1 MW and more) ORC systems powered by the heat sources with high thermal power and temperature (150 °C and more). Such heat sources are generated as waste heat in large industrial power machines e.g. steam boilers (waste steam) or gas turbines (exhaust gases). Lai et. al. (2011) showed that in the large power systems silicone oils (e.g. MM (hexamethyldisiloxane) or MDM (octamethyltrisoloxane)) are mainly adopted as working fluids.

Gnutek and Kolasiński (2013) indicated that volumetric expanders are applicable mainly in micro and small power systems such as domestic and agriculture plants powered by the heat sources with small capacities, thermal power and temperature (up to 150 °C). Low thermal parameters of heat source influences also the working fluid selection. Only the low-boiling working fluids are possible for application in this case. Such working fluid are the refrigerants and similar substances e.g. classical R123 ($C_2HCl_2F_3$) and R245fa ($C_3H_3F_5$), as well as new specially designed fluids e.g. R1234yf ($C_3F_4H_2$), R1234ze ($C_3F_4H_2$), or SES36 ($CF_3CH_2CF_2CH_3$/PFPE).
Volumetric expanders are a good option for systems where the low pressures and low working medium flows are expected. In general piston, screw, spiral, vane and rotary lobe expanders can be applied in ORC plants. Piston expanders are good option for the ORC systems were high (up to 20 MPa in case of the single stage expanders) inlet pressures of the working fluid are expected, as they have the high expansion ratios (the ratio of the inlet and outlet pressure). Piston expanders can be used in the ORC systems powered by heat sources with stable characteristic of the thermal power output as these expanders must work in dry vapor conditions in order to avoid liquid phase in the cylinder. Screw expanders can be applied in systems powered by heat sources with changeable characteristic (both in terms of the temperature, capacity and power) as in this type of the expanders moist vapor can be expanded without problems. The expansion ratio of the screw expander typically is in the range of σ = 10—15. Spiral expanders are applied in many of the ORC systems as they are compact and relatively cheap. Vane expanders are applied mostly in the ORC prototypes and test-stands and most of them is under research and development. The expansion ratio of the rotary vane expanders typically is in the range of σ = 5—7. Rotary lobe expanders are also under research and development, but these type of the expanders are promising because of their advantages such as e.g. ability to expand the moist vapor, low operating pressures and simple design. The range of working pressures and expansion ratios of different types of volumetric expanders are presented in table 1.

Table 1: Range of the working pressures and expansion ratios for different types of the volumetric expanders (Więckiewicz and Cantek (1985))

Expander type	$p_{in\ max}$	p_{out}	σ_{max}
	MPa	**MPa**	**-**
Piston (single stage)	20	0.1	200
Screw	1.5	0.1	15
Spiral	1.0	0.1	10
Vane	0.7	0.1	7
Rotary lobe	0.6	0.1	6

The most important feature of the volumetric expander is the relationship of the expander power and the expansion ratio. This issue was discussed in details by Gnutek and Kolasiński (2011). Each type of volumetric expander also has the optimum value of the expansion ratio. Unlike the turbines, volumetric expanders can operate at low working fluid flow rates and low pressures. The task of a suitable working fluid selection to the ORC system with volumetric expander should be thus considered differently than in the case of the turbine-based systems.

2. THERMODYNAMIC PROPERTIES OF WORKING FLUIDS SUITABLE FOR ORC SYSTEMS WITH VOLUMETRIC EXPANDERS

As it was indicated in the introduction volumetric expanders are good option for ORC systems powered by low temperature heat sources and each type of the volumetric expander has specific range of the expansion ratio. Taking into account this specific conditions (low temperature of the heat source and expander operational conditions) only the selected working fluids can be applied in ORC system with volumetric expanders. Selected working fluids that can be used in low-power ORC systems with volumetric expanders are presented in table 2.

Working fluids can be described by well-known thermodynamic relationships (equations of the state, the expressions for the specific heat, heat of a phase change or the thermodynamic functions), and the coefficients defined for the description and analysis of specific applications. The above-mentioned thermodynamic relations are presented in the form of algebraic or differential equations, tables, graphs or software. Level of completeness of this description in relation to the substances, presented in table 2, is very different, which does not facilitate the thermodynamic analyzes. Figure 1 shows the general T-s diagram for the low-boiling substance. The characteristic values of thermal properties are indicated on this graph with taking into account the ambient parameters. The areas of the individual phases can therefore be highlighted: superheated vapor, moist vapor, liquid, the area of the critical point (C_r), the dry saturated vapor line (x = 1) and the line of boiling liquid (x = 0). Working fluid with ambient temperature T_a has the pressure $p(T_a)$, typically different from the ambient pressure p_a. Evaporation temperature T_{ev} at ambient pressure (p_a) is one of the basic quantities describing the substance, just like the corresponding heat of condensation $q_c(p_a)$ (isobar, isotherm e-f). The critical point parameters – p_{cr}, T_{cr} are another key parameters describing the substance. Isobar passing through this point is important in the division of the operating range of the power plant and organization of the cycle. In contrast, the critical isotherm determines the usefulness of a substance to act as a heat transfer fluid in a power plant. If $T_c < T_a$ it would be not possible to condense the vapor and liquid compression, which is the basic principle of a power plant operation. On the background of the described T-s graph the power plant cycle was presented (red lines).

Table 2: The working fluids suitable for low-power ORC systems with volumetric expanders

No.	**Margin Position**	**Chemical formula**	**Molar mass**	**Normal boiling point**	**Critical point**			**Range of applicability**		
			M	t_n	t_{cr}	p_{cr}	ρ_{cr}	t_{min}	t_{max}	p_{max}
			kg/kmol	**°C**	**°C**	**MPa**	**kg/m^3**	**°C**	**°C**	**MPa**
1	R113	CCl_2FCClF_2	187.38	47.5	214.06	3.39	560.00	-36.22	251.85	200
2	R114	$C_2Cl_2F_4$	170.92	3.6	145.68	3.25	579.97	0	233.85	21
3	R123	$CHCl_2CF_3$	152.90	27.8	183.68	3.66	550.00	-107.15	326.85	40
4	R124	CF_3CHClF	136.50	-11.9	122.28	3.62	560.00	-153.15	196.85	40
5	R1234ze	$CHF{=}CHCF_3$	114.04	-18.9	109.37	3.63	489.24	-104.53	146.85	20
6	R134a	$CH_2F\text{-}CF_3$	102.00	-26.0	101.06	4.06	512.00	-103.3	181.85	70
7	R152a	CHF_2CH_3	66.05	-24.0	113.26	4.51	368.00	-118.59	226.85	60
8	R227	CF_3CHFCF_3	170.03	-16.3	101.75	2.93	594.25	-126.8	201.85	60
9	R236fa	$CF_3CH_2CF_3$	152.00	-1.4	124.92	3.20	551.30	-93.63	226.85	40
10	R245fa	$CF_3CH_2CHF_2$	134.05	15.1	154.01	3.65	516.08	-102.1	166.85	200
11	R365mfc	$CF_3CH_2CF_2CH_3$	148.07	40.1	186.85	3.22	473.84	-34.15	226.85	35
12	R409A	-	97.40	-34.2	107.00	4.60	-	-	-	-
13	SES 36	$CF_3CH_2CF_2CH_3$/PFPE	184.85	35.6	177.60	2.85	-	-	-	-

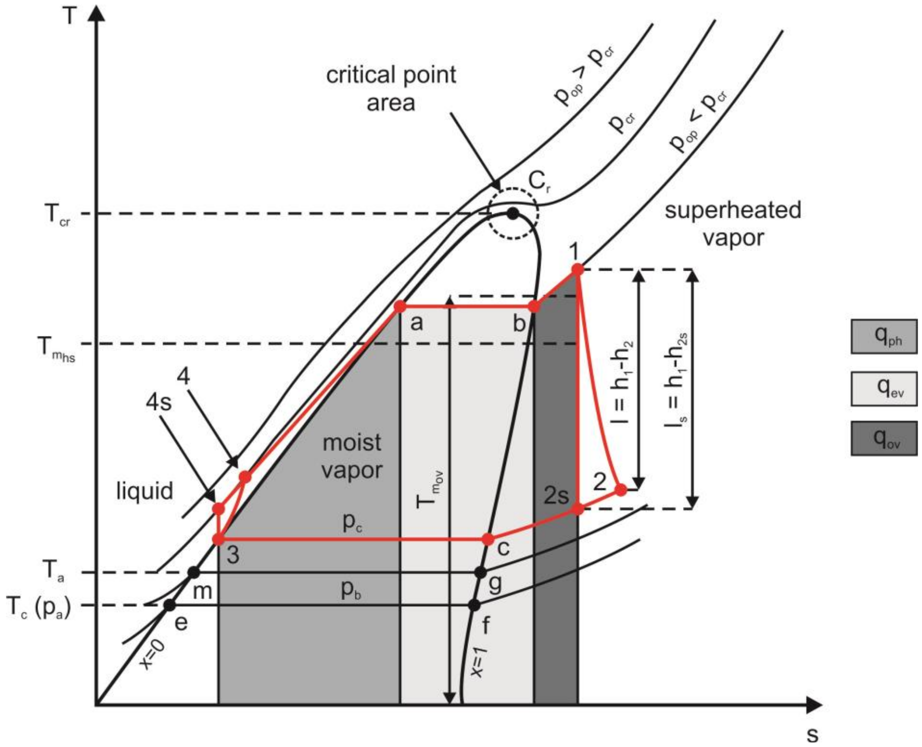

Figure 1: T-s diagram for low-boiling working fluid

Depending on the working fluid type the characteristic curve can have different shapes. Figure 2 shows the characteristic curves for selected working fluids. Also the temperature range for low potential heat sources is presented on this figure with dashed lines (90—150 °C).

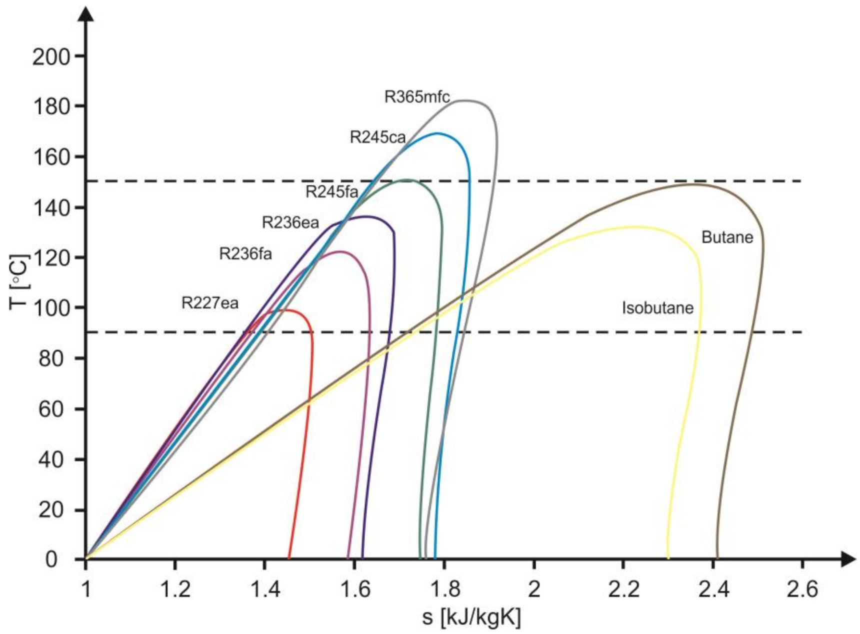

Figure 2: Characteristic curves for different working fluids (Trapp and Colonna (2013))

3. THE METHOD OF WORKING FLUID SELECTION

Figure 3 shows the comparison of two ideal power plant cycles (blue and red lines) for two different low-boiling working fluids on T-s plane, which is the basis for further considerations. This graph is built with the assumption that both of the cycles are powered by the same heat source with the same heat supply temperature (T_{hs}). Moreover it is assumed that both of the cycles are cooled by the same heat sink with the same condensation temperature (T_c).

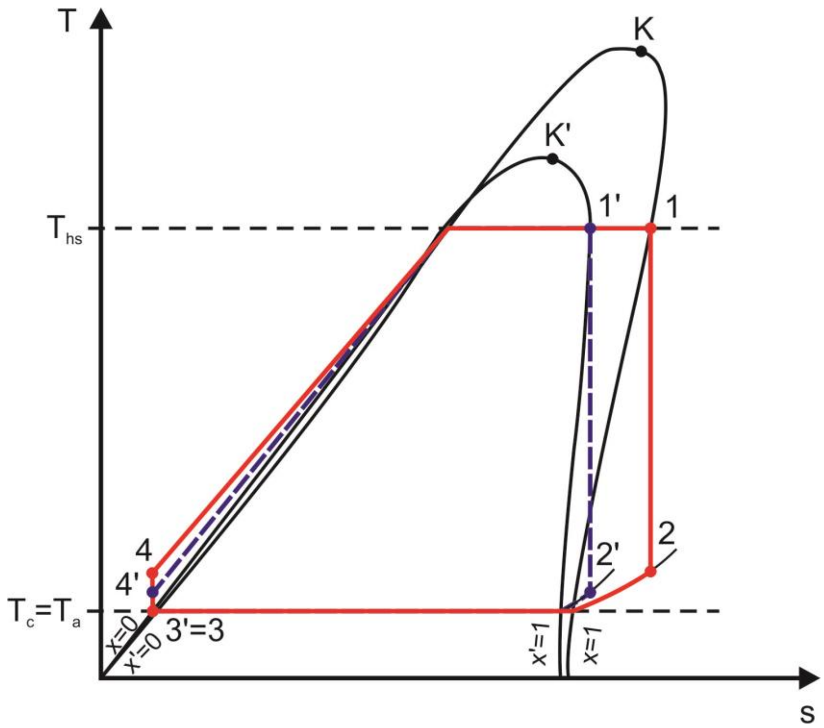

Figure 3: T-s diagram with the power plant cycles for two different working fluids

Comparison of two different substances is possible basing on many parameters (e.g. toxicity, ODP, HGWP, etc.), but the most important are the thermal properties as they have the influence on the ORC system power, efficiency and quality of operation. Thus, in this methodology, it is proposed to analyse the substances basing on the following thermal properties:

- The amount of the heat absorbed from the heat source in the evaporator,
- The mean temperature of the heat supply,
- The amount of the heat removed by the heat sink in the condenser,
- The efficiency of thermal energy conversion.

Moreover the following assumptions are needed to carry out the comparison (see Fig. 3 for details):

- The temperatures of the heat source and the heat sink are the same for both substances,
- Both working substances are thermally stable in the described on Fig. 3 temperature range,
- One of the substances is the reference.

In order to compare two substances with taking into account above mentioned properties and assumptions it is useful to define dimensionless comparative parameters, described in the following.

3.1 α_{hs} - The parameter of the absorbed heat

α_{hs} parameter characterizes the working fluid in terms of ability to absorb heat at the possible highest supply temperature (T_{hs}). This parameter can be useful when comparing the working fluids in terms of maximizing the heat amount absorbed from the heat source. The comparison of α_{hs} parameter calculated for two different working fluids and two different ORC cycles (the reference one and compared one, as presented on Fig.3) allows the selection of a working fluid with larger heat absorption capacity, i.e. one that will maximize the heat absorption from the heat source. This parameter can be defined with the expression

$$\alpha_{hs} = \left(\frac{\dot{Q}^{+}_{ORC1}}{\dot{Q}^{+}_{ORC2}} \right)_{(T_{hs}-T_a),\,p} \tag{1}$$

$$\dot{Q}^{+}_{ORC1} = \dot{m}_{ORC1} \cdot \left(h_1 - h_4 \right) \tag{2}$$

$$\dot{Q}^{+}_{ORC2} = \dot{m}_{ORC2} \cdot \left(h_{1'} - h_{4'} \right) \tag{3}$$

Where:

$\dot{m}_{ORC1}$ - the working medium mass flow in the compared cycle,

$\dot{m}_{ORC2}$ - the working medium mass flow in the reference cycle,

$\dot{Q}^{+}_{ORC1}$ - the heat transfer rate absorbed by the working fluid during the evaporation in the compared cycle,

$\dot{Q}^{+}_{ORC2}$ - the heat transfer rate absorbed by the working fluid during the evaporation in the reference cycle.

3.2 α_{mhs} - The parameter of the mean heat supply temperature

α_{mhs} parameter characterizes the working fluid in terms of ability to absorb heat at the highest possible mean heat delivery temperature T_{mhs}. This parameter can be useful for comparing the working fluids in terms of maximizing the heat delivery temperature and comparative analyses of working fluids in terms of maximizing ORC plant efficiency. This parameter can be defined with the expression

$$\alpha_{m_{hs}} = \left(\frac{T^{mhs}_{ORC1}}{T^{mhs}_{ORC2}} \right)_{(T_{hs}-T_a),\,p} \tag{4}$$

$$T^{mhs}_{ORC1} = \frac{q_{ph} + q_{ev} + q_{sh}}{\Delta s} \tag{5}$$

$$T^{mhs}_{ORC2} = \frac{q_{ph}' + q_{ev}' + q_{sh}'}{\Delta s'} \tag{6}$$

Where:

T^{mhs}_{ORC1} - mean temperature of the heat supply in the compared cycle,

T^{mhs}_{ORC2} - mean temperature of the heat supply in the reference cycle,

q_{ph} - heat of preheating,

q_{ev} - heat of evaporation,

q_{sh} - heat of superheating,

Δs - change in the medium entropy during the preheating, evaporation and superheating.

3.3 α_o - The parameter of the removed heat

α_o parameter characterizes the working fluid in terms of ability to remove the heat at the lowest possible heat removal temperature $T_c \cong T_a$. This parameter can be useful for comparing the working fluids in terms of minimizing the temperature of the heat sink. This parameter can be defined as:

$$\alpha_{hr} = \left(\frac{\dot{Q}^{-}_{ORC1}}{\dot{Q}^{-}_{ORC2}} \right)_{(T_{hs}-T_a),\, p} \tag{7}$$

$$Q^{-}_{ORC1} = \dot{m}_{ORC1} \cdot \left(h_{ORC1}'' - h_{ORC1}' \right) \tag{8}$$

$$Q^{-}_{ORC2} = \dot{m}_{ORC2} \cdot \left(h_{ORC2}'' - h_{ORC2}' \right) \tag{9}$$

Where:

$\dot{m}_{ORC1}$ - the working medium mass flow in the compared cycle,

$\dot{m}_{ORC2}$ - the working medium mass flow in the reference cycle,

$\dot{Q}^{-}_{ORC1}$ - the heat transfer rate removed during the condensation in the compared cycle,

$\dot{Q}^{-}_{ORC2}$ - the heat transfer rate removed during the condensation in the reference cycle.

3.4 α_e – The parameter of the efficiency of thermal energy conversion

α_e parameter characterizes the working fluid in terms of ability to maximize the efficiency of the energy conversion. This parameter can be useful when comparing the working fluids in terms of maximizing the efficiency of conversion of the heat supplied from the heat source to the other energy forms. This parameter can be defined as:

$$\alpha_e = \left(\frac{E_{ORC1}}{E_{ORC2}} \right)_{(T_{hs}-T_a),\, p} \tag{10}$$

$$E_{ORC1} = P_{ORC1} + Q_{T\,ORC1} + Q_{L\,ORC1} \tag{11}$$

$$E_{ORC2} = P_{ORC2} + Q_{T\,ORC2} + Q_{L\,ORC2} \tag{12}$$

Where:

E_{ORC1} - the energy generated and dissipated in the compared cycle,

E_{ORC2} - the energy generated and dissipated in the reference cycle,

P - electric power output (the electric power on the output of the current generator),

Q_T - thermal power output (heat generated for the central heating purposes),

Q_L - heat losses (heat losses via convection and radiation from the surfaces of the pipelines, the evaporator, the condenser and the other system devices).

The above defined parameters can be useful for comparative selection of the ORC system working fluid, when the thermal characteristic of the heat source and the heat sink (namely the heat source and the heat sink temperatures) are known. The selection criteria are the maximal values of the α_{hs}, α_{mhs}, α_{hr} and α_e parameters. Moreover, basing on the comparison of the calculated parameters and the values of the expansion ratio as well as specific expansion work it is possible to select a suitable volumetric expander to the ORC system. The importance of the α_{hs}, α_{mhs}, α_{hr} and α_e parameters is always connected with the application of the ORC system. For the example in the ORC system dedicated for heat recovery from the cooling mediums (e.g. in internal combustion engines) the heat removal from the medium is the priority. Thus, the most important parameter in this case is the α_{hs} and it should be maximized during the expander and working fluid selection. In each ORC system cycle efficiency is also very important, thus in all of the applications α_e parameter should be maximized.

4. COMPARATIVE ANALYSIS OF THE WORKING FLUIDS

In the following the comparative analysis of the selected low-boiling working fluids suitable for application in ORC powered by low temperature heat sources is presented. The analysis was performed for working fluids listed in table 2. The values of thermal properties (i.e. specific enthalpy and the specific entropy) were calculated with the Refprop and Solkane software.
The following assumptions were taken into account in the calculations:

- The temperature of the heat source is t_{hs} = 95 °C,
- The expander internal efficiency is η_i = 0,7,
- Temperature of the heat sink is t_c = 20 °C,
- The reference substance is R123,
- There is no heat generated for central heating and heat losses in the system are negligible.

The defined earlier comparative parameters (α_{hs}, α_{mhs}, α_{hr} and α_e) together with the expansion ratio and expander specific work were calculated for each of the working fluid. Calculations were made for cycles presented in fig. 3. The results of these calculations are presented in table 3 (indexes of the thermal properties according to fig. 3).

Table 3: The results of the calculations

Working fluid	t_1	p_1	t_2	p_2	h_1	s_1	i_2	s_2	h_3'	s_3'	l	σ	α_{hs}	α_{mhs}	α_{hr}	α_e
	°C	bar	°C	bar	kJ/kg	kJ/kgK	kJ/kg	kJ/kgK	kJ/kg	kJ/kgK	kJ/kg	-	-	-	-	-
R113	95	3.87	50.26	0.37	417.07	1.6262	391.75	1.6527	218.09	1.0639	25.32	10.55	0.91	1.00	0.91	0.92
R114	95	12.83	48.42	1.82	390.81	1.5562	369.83	1.5783	219.44	1.0684	20.98	7.05	0.79	0.99	0.79	0.76
R123	95	7.02	42.45	0.76	438.19	1.6890	410.56	1.7188	220.05	1.0763	27.63	9.24	1.00	1.00	1.00	1.00
R124	95	21.54	23.65	3.27	404.41	1.5971	381.89	1.6220	222.09	1.0779	22.52	6.59	0.84	0.99	0.84	0.82
R1234ze	95	27.39	28.02	4.27	252.31	0.7610	229.22	0.7931	51.019	0.1856	23.09	6.41	0.92	0.98	0.94	0.84
R134a	95	35.91	20.00	5.72	421.12	1.6504	397.73	1.6770	227.47	1.0960	23.39	6.28	0.89	0.98	0.89	0.85
R152a	95	31.79	20.00	5.13	540.50	2.3412	501.80	2.3852	234.77	1.4755	38.70	6.20	1.40	0.99	1.40	1.40
R227	95	25.50	37.64	3.90	369.04	1.5025	352.37	1.5206	222.81	1.0806	16.67	6.54	0.67	0.98	0.68	0.60
R236fa	95	17.44	43.36	2.30	416.85	1.6370	393.91	1.6615	224.62	1.0867	22.94	7.60	0.88	0.99	0.89	0.83
R245fa	95	11.30	43.05	1.22	471.29	1.7885	440.70	1.8213	225.86	1.0912	30.59	9.21	1.13	0.99	1.13	1.11
R365mfc	95	5.20	56.16	0.46	492.00	1.8487	459.26	1.8824	226.90	1.0956	32.74	11.30	1.22	0.99	1.22	1.18
R409A	95	35.99	19.33	5.78	412.28	1.6186	387.37	1.6474	215.20	1.0495	24.91	6.23	0.90	0.97	0.90	0.90
SES 36	95	5.61	60.64	0.58	414.77	1.6176	390.30	1.6423	220.28	1.0705	24.47	9.67	0.89	1.00	0.89	0.89

As it can be seen from the calculation results, three of the working fluids (R152a, R245fa and R365mfc) have better thermal properties in comparison to R123 as the comparative parameters are greater than 1.
For the assumed ORC system working conditions, the best working fluid is R152a when compared to R123. In case of R152a the calculated values of the α_{hs}, α_{hr} and α_e are equal (1.4). This means that R152a is 40% better than R123 in the heat absorption, heat removal and thermal energy usage efficiency. The α_{mhs} parameter is lower (α_{mhs} = 0.99) in case of R152a when compared to R123, thus the mean heat supply temperature is lower for R152a. Also, the value of the expansion ratio is lowest (σ = 6.2) in case of R152a in comparison to other analyzed substances, and the specific expansion work is highest (l = 38.7 kJ/kg). However, in case of R152a the pressure on the expander inlet is high (p_1 = 31.79 bar) and the choice of the suitable volumetric expander is limited only to the piston expander (see Table 1 for maximum expansion ratio of different volumetric expanders).

The second, better than R123 working fluid, is R365mfc. In case of R365mfc the calculated values of the α_{hs}, α_{hr} and α_e are 1.22, 1.22 and 1.18 correspondingly. This means that R365mfc is 22% better than R123 in the heat absorption and heat removal and 18% better in the thermal energy usage efficiency. Similarly to the earlier described comparison α_{mhs} parameter is also lower (α_{mhs} = 0.99) in this case. The value of the expansion ratio in case of R365mfc is higher than in the case of R123 (σ = 11.3), but the operational pressures are lower (p_1 = 5.20 bar and p_2 = 0.46 bar) and the specific expansion work is higher (l = 32.74 kJ/kg) when compared to R123. Low operational pressures and the value of the expansion ratio makes that screw expander will be optimal in this case.
The third, better than R123, working fluid is R245fa. In case of R245fa the calculated values of the α_{hs}, α_{hr} and α_e are 1.13, 1.13 and 1.11 correspondingly. This means that R245fa is 13% better than R123 in the heat absorption and heat removal and 11% better in the thermal energy usage efficiency. Similarly to the earlier described comparisons α_{mhs} parameter is also lower (α_{mhs} = 0.99) in this case. The value of the expansion ratio in case of R245fa is lower than in the case of R123 (σ = 9.21), but the operational pressures are higher (p_1 = 11.30 bar and p_2 = 1.22 bar). The specific expansion work is higher (l = 30.59 kJ/kg) when compared to R123. The moderate pressures and the value of the expansion ratio in this case makes that two types of the volumetric expanders (screw and spiral) can be applied in this case. As it can be seen from the results α_{hs} and α_{hr} are similar for the analyzed substances. This is due to the similar shapes of the saturation curves for these substances and similar ratio of the heat needed for preheating, evaporation and superheating of the substance in relation to the heat of condensation. The other working fluids (R113, R114, R124, R1234ze, R134a, R227, R236fa, R409A and SES 36) have worse thermal properties when compared with R123. The calculated comparative parameters are lower for these substances in comparison to R123.

5. CONCLUSIONS

This study presents the comparative working fluid selection method for ORC system powered by low temperature heat source. This method is based on comparison of selected thermal properties of working fluids with the use of the defined parameters describing the substance in the following properties: the amount of the heat absorbed from the heat source, the mean temperature of the heat supply, the amount of the heat removed by the heat sink and the efficiency of thermal energy conversion. Described method can be useful for the working fluid selection to the ORC system with known thermal parameters (temperatures) of the heat source and the heat sink. Moreover, it is possible to select the suitable volumetric expander basing on the comparison of the calculated parameters, the expansion ratio and the specific expansion work. The comparison example presented in point 3 of this paper and valid for the ORC system powered by the low-temperature (95 °C) heat source, where R123 is the reference working fluid, shows that in considered case three different working fluids (R152a, R245fa and R365mfc) can be more efficient alternatives to R123. Also, the suitable volumetric expanders were selected using the presented method for each of the working fluids alternatives. Presented method can also easily be adopted to other working substances and other assumptions. Thus, using this method, it is possible to analyze many different ORCs powered by different heat sources.

NOMENCLATURE

The nomenclature should be located at the end of the text using the following format:

α	comparative parameter	(–)
η	efficiency	(–)
ρ	density	(kg/m^3)
σ	expansion ratio	(–)
E	energy	(J)
h	specific enthalpy	(kJ/kg)
l	specific work	(kJ/kg)
M	molar mass	(kg/kmol)

$\dot{m}$	mass flow	(kg/s)
P	power	(W)
p	pressure	(Pa)
r	heat of evaporation	(kJ/kg)
q	specific heat	(kJ/kg)
$\dot{Q}$	heat transfer rate	(W)
s	specific entropy	(kJ/kgK)
T	temperature	(K)
t	temperature	(°C)
x	vapor quality	(–)

Subscript

1, 2, …, n	1st, 2nd, … nth
a	ambient
c	condensation
cr	critical
e	efficiency
ev	evaporation
hs	heat supply
i	internal
in	inlet
L	losses
m	mean
max	maximal
min	minimal
out	outlet
op	operational
ORC1	related to the compared cycle
ORC2	related to the reference cycle
sh	superheating
ph	preheating
s	isentropic
T	thermal

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