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EVAPORATOR FOR ORC CYCLE WITH RECIRCULATING HEAT CARRING WATER – COMPUTATIONAL MODEL

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Abstract: The following paper presents an ORC installation including an evaporator with recirculation (heat carrying water exiting the evaporator is redirected to its inlet). It covers the calculations of inlet/outlet temperature of the evaporator taking into account a variant recirculation coefficient. Formulas for heat transfer between heat carrying water and working fluid inside evaporator are also included in this paper. The calculations are based on properly defined average specific heat. The analysis shows that the system performance depends on heat carrying water inlet temperature, on heat carrying water flow rate and the recirculation coefficient.

Keywords: evaporator, recirculation, average specific heat, ORC power plant, mathematical modeling

1.INTRODUCTION

This paper presents a computational model for a power plant with single subcritical ORC loop (Fig. 1) [10-12] that instead of a classic evaporator includes one with recirculating heat carrying water. The computational model is based on a general steady state energy balance

$$\dot{E}_d = \dot{E}_w.$$

In order to obtain a simple computational model, two types of average specific heat were used: $c\big|_0^T$ and $c\big|_{T_2}^{T_1}$. The first average specific heat $c\big|_0^T$ was used to determine the specific enthalpy of heat carrying water

$$h = c\big|_0^T T$$

that allows to calculate the enthalpy of water flux using the following formula:

$$\dot{H} = \dot{m} \cdot h.$$

A relation between two average specific heats $c\big|_0^{T_1}$ and $c\big|_0^{T_2}$ was used to derive $c\big|_{T_2}^{T_1}$.

The computational model [12] is based on an assumptions that the energy is supplied to the system by hot water with mass flow rate $\dot{m}_s$ and temperature T_{s1}.

The main purpose of the formulas within the computational model is to estimate the evaporator inlet and outlet temperatures T_{s1}^z and T_{s2}^z taking into account temperature T_{s1}, recirculation coefficient z and evaporator temperature difference

$$\Delta T_s^z = = T_{s1}^z - T_{s2}^z.$$

The analysis conducted proves that the evaporator inlet and outlet temperatures can be calculated using the following formulas:

$$T_{s1}^z = f\left(T_{s1}, z, \Delta T_s^z\right) \text{ and } T_{s2}^z = f\left(T_{s1}, z, \Delta T_s^z\right).$$

The analysis also shows that the heat $\dot{Q}_s$ transferred from water to working fluid, basing on the energy balance, can be calculated according to the following formula:

$$\dot{Q}_s = (1+z)\dot{m}_s \bar{c}_s \left(T_{s1}^z - T_{s2}^z\right) = = \dot{m}_n \Delta h_{par}.$$

The mass flow rate of heat carrying water $\dot{m}_{s1}$ suppling the counter current working fluid pre-heater can be calculated using energy balance of this heat exchanger and average specific heat $\bar{c}_{s2}$.

Fig. 1. ORC power plant supplied with water at 100°C (Department of Heat Engineering, West Pomeranian University of Technology in Szczecin, operating factor R227ea, 9 kW$_{el}$ power) [11]

2.ORC POWER PLANT WITH RECIRCULATING HEAT CARRYING WATER

In a power plant with single ORC loop it is possible to select a proper working fluid. Currently, there is a variety of substances that can serve as working fluid: natural pure substances (butane, izobutane, propane), synthetic pure substances and blends [1-8, 14-16]. Utilization of organic substance allows to achieve better working conditions comparing to the water (for the same pressure and temperature). The main advantage of those substances over the water is low evaporation temperature (for some of them even below 100°C).

A scheme of a single loop ORC power plant with recirculating heat carrying water evaporator is presented in Fig. 2.

The principle of operation for ORC power plant with recirculating heat carrying water evaporator is simple. First, the heat carrying water is directed to the evaporator. Due to recirculation, flow rate through evaporator (node A) is a sum of circulating water $\dot{m}_s$ and the recirculating water $\Delta\dot{m}_s$. Thus, the temperature at evaporator inlet drops to T_{s1}^z. The flow rate at the evaporator outlet ($\dot{m}_s + \Delta\dot{m}_s$) splits into two streams (node B). The $\Delta\dot{m}_s$ stream is redirected through by-pass to the evaporator inlet where it meets heat carrying water, and the $\dot{m}_s$ stream goes further into the system. Then the $\dot{m}_s$ stream splits into two streams $\dot{m}_{s1}$ and $\dot{m}_{s2}$: the $\dot{m}_{s1}$ stream is directed to ORC working fluid pre-heater ($\dot{m}_{s1}$ can be determined using energy balance equation for the pre-heater); the $\dot{m}_{s2}$ stream can be calculated according to the formula $\dot{m}_{s2} = \dot{m}_s - \dot{m}_{s1}$, at temperature T_{s2}^z it reconnects the $\dot{m}_{s1}$ stream at temperature T_{s3} at node C. The mixed stream with flow rate $\dot{m}_s = \dot{m}_{s1} + \dot{m}_{s1}$ and temperature T_{s4} is directed to a heat exchanger where the temperature rises from T_{s4} to T_{s1}. On the secondary side of the heat exchanger there is a waste heat carrier with flow rate $\dot{m}_o$ that operates in a temperature range between T_{o1} and T_{o2}.

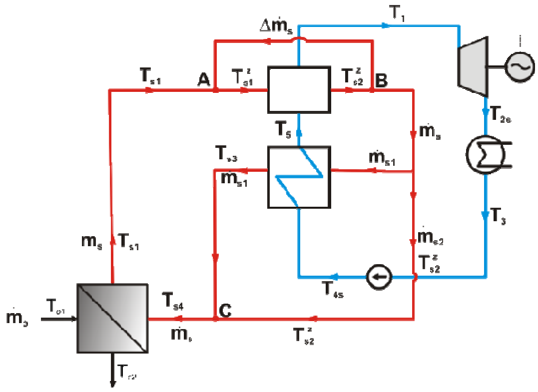

Fig. 2. A scheme of single loop ORC power plant with recirculating heat carrying water evaporator [12]

3.A METHOD FOR MODELLING OF RECIRCULATING HEAT CARRYING WATER EVAPORATOR

Fig. 3 shows a scheme of recirculating heat carrying water evaporator. It includes three balance sections (space inside evaporator occupied by liquid **a** and space occupied **c** occupied by evaporating working fluid) together with all important parameters necessary for computation.

The energy balance equation for water space inside the evaporator, including the heat flux $\dot{Q}_s$

supplied to the evaporating medium (balance section **a** in Fig. 3) looks as follow:

$$(1+z)\dot{m}_s \cdot c\Big|_0^{T_{s1}^z} T_{s1}^z = (1+z)\dot{m}_s \cdot c\Big|_0^{T_{s2}^z} T_{s2}^z + \dot{Q}_s \quad (1)$$

or after rearrangement:

$$\dot{Q}_s = (1+z)\dot{m}_s \left[c\Big|_0^{T_{s1}^z} T_{s1}^z - c\Big|_0^{T_{s2}^z} T_{s2}^z \right]. \quad (1a)$$

Next, the average specific heat is introduced $\bar{c}_s = c\Big|_{T_{s2}^z}^{T_{s1}^z}$.

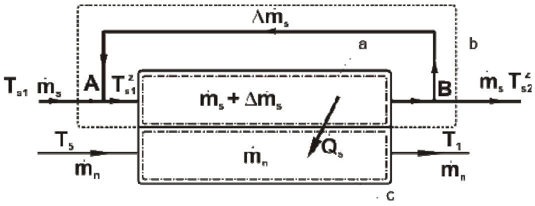

Fig. 3. Scheme of recirculating heat carrying water evaporator [12]

This heat is defined as a function of two temperatures of recirculating water at evaporator inlet and outlet, T_{s1}^z and T_{s2}^z, respectively, and two average specific heats $c\Big|_0^{T_{s1}^z}$ and $c\Big|_0^{T_{s2}^z}$:

$$\bar{c}_s = c\Big|_{T_{s2}^z}^{T_{s1}^z} = \frac{c\Big|_0^{T_{s1}^z} T_1^z - c\Big|_0^{T_{s2}^z} T_{s2}^z}{T_{s1}^z - T_{s2}^z}. \quad (2)$$

After substituting (1a) into (2):

$$\bar{c}_s \left(T_{s1}^z - T_{s2}^z \right) = c\Big|_0^{T_{s1}^z} T_{s1}^z - c\Big|_0^{T_{s2}^z} T_{s2}^z, \quad (2a)$$

after simple rearrangements, the heat flux from water to evaporating working medium can be written in following form:

$$\dot{Q}_s = (1+z)\dot{m}_s \cdot \bar{c}_s \left(T_{s1}^z - T_{s2}^z \right). \quad (3)$$

In order to determine the recirculating water temperature, the energy balance equation for node A is used:

$$\dot{m}_s \cdot c\Big|_0^{T_{s1}} T_{s1} + z\,\dot{m}_s \cdot c\Big|_0^{T_{s2}^z} T_{s2}^z = (1+z)\dot{m}_s \cdot c\Big|_0^{T_{s1}^z} T_{s1}^z \quad (4)$$

or after rearrangements the formula (4) takes the following form:

$$c\Big|_0^{T_{s1}} T_{s1} - c\Big|_0^{T_{s1}^z} T_{s1}^z = z\left(c\Big|_0^{T_{s1}^z} T_{s1}^z - c\Big|_0^{T_{s2}^z} T_{s2}^z \right). \quad (4a)$$

After introducing another average specific heat $\bar{c}_{s1} = c\Big|_{T_{s1}^z}^{T_{s1}}$ defined as follow:

$$\bar{c}_{s1} = c\Big|_{T_{s1}^z}^{T_{s1}} = \frac{c\Big|_0^{T_{s1}} T_{s1} - c\Big|_0^{T_{s1}^z} T_{s1}^z}{T_{s1} - T_{s1}^z} \quad (5)$$

and average specific heat $\bar{c}_s$ from (2a) into (4a), it can be rewritten to the following form:

$$\bar{c}_{s1}\left(T_{s1} - T_{s1}^z \right) = z \cdot \bar{c}_s \left(T_{s1}^z - T_{s2}^z \right). \quad (4b)$$

After proper rearrangements, the temperature or recirculating water can be calculated according to the following formula:

$$T_{s1}^z = T_{s1} - z \cdot \frac{\bar{c}_s}{\bar{c}_{s1}} \Delta T_s^z. \quad (6)$$

To derive the formula for water temperature at evaporator outlet, it is necessary to make an energy balance for section **b** (Fig. 3):

$$\dot{m}_s \cdot c\Big|_0^{T_{s1}} T_{s1} = \dot{Q}_s + \dot{m}_s \cdot c\Big|_0^{T_{s2}} T_{s2}, \quad (7)$$

that after substituting (3) takes the following form:

$$c\Big|_0^{T_{s1}} T_{s1} - c\Big|_0^{T_{s2}} T_{s2} = (1+z)\bar{c}_s \left[T_{s1}^z - T_{s2}^z \right]. \quad (7a)$$

After using the definition of average specific heat $\bar{c}_{s2} = c\Big|_{T_{s2}^z}^{T_{s1}}$:

$$\bar{c}_{s2} = c_{s2}\Big|_{T_{s1}^z}^{T_{s1}} = \frac{c\Big|_0^{T_{s1}} T_{s1} - c\Big|_0^{T_{s2}^z} T_{s2}^z}{T_{s1} - T_{s2}^z} \quad (8)$$

the (7a) can be transformed into a formula for water temperature T_{s2}^z :

$$\bar{c}_{s2}\left(T_{s1} - T_{s2}^z \right) = (1+z)\bar{c}_s \left(T_{s1}^z - T_{s2}^z \right) \quad (9)$$

or:

$$T_{s2}^z = T_{s1} - (1+z) \cdot \frac{\bar{c}_s}{\bar{c}_{s2}} \Delta T_s^z. \quad (9a)$$

Using the formulas (6) and (9a) that allow to calculate water temperature at evaporator inlet and outlet, T_{s1}^z and T_{s2}^z, respectively, it is possible to evaluate the temperature difference T_{s2}^z :

$$T_{s1}^z - T_{s2}^z = \beta \Delta T_s^z, \quad (10)$$

where: $$\beta = \frac{\bar{c}_s}{\bar{c}_{s2}} + z\left(\frac{\bar{c}_s}{\bar{c}_{s2}} - \frac{\bar{c}_s}{\bar{c}_{s1}} \right). \quad (11)$$

The numerical values of average specific heat $\bar{c}_s$, $\bar{c}_{s1}$ and $\bar{c}_{s2}$ were calculated using specific enthalpy h' for saturated water available in REFPROP 9 software [9]:

$$\bar{c}_s = \frac{h'\left(T_{s1}^z\right) - h'\left(T_{s2}^z\right)}{T_{s1}^z - T_{s2}^z} =$$

$$= \frac{h'\left(T_{s1} - z\Delta T_s^z\right) - h'\left[T_{s1} - (1+z)\Delta T_s^z\right]}{\Delta T_s^z},$$

$$\bar{c}_{s1} = \frac{h'\left(T_{s1}\right) - h'\left(T_{s1}^z\right)}{T_{s1} - T_{s1}^z} = \frac{h'\left(T_{s1}\right) - h'\left(T_{s1} - z\Delta T_s^z\right)}{z\Delta T_s^z},$$

for $z \neq 0$,

$$\bar{c}_{s2} = \frac{h'\left(T_{s1}\right) - h'\left(T_{s2}^z\right)}{T_{s1} - T_{s2}^z} = \frac{h'\left(T_{s1}\right) - h'\left[T_{s1} - (1+z)\Delta T_s^z\right]}{(1+z)\Delta T_s^z}.$$

The results of average specific heat calculations are presented in tables 1-6. They served as a base for average specific heat charts $\bar{c}_s = f(T_{s1}, z)$, $\bar{c}_{s1} = f(T_{s1}, z)$, $\bar{c}_{s2} = f(T_{s1}, z)$ (Fig. 4-6) and ratios of average specific heat chart $\bar{c}_s/\bar{c}_{s1} = f(T_{s1}, z)$, $\bar{c}_s/\bar{c}_{s2} = f(T_{s1}, z)$ and $\beta = f(T_{s1}, z)$ – Fig. 7-9.

The analysis conducted, for the water temperature range $T_{s1} = 220 \div 90°C$ and recirculation coefficient range $z = 0 \div 12$, shows that $\bar{c}_s/\bar{c}_{s1} = f(T_{s1}, z) \cong 1$; $\bar{c}_s/\bar{c}_{s2} = f(T_{s1}, z) \cong 1$ and $\beta = f(T_{s1}, z) \cong 1$.

Taking the above into consideration, basing on formula (6), the temperature of recirculating water at the evaporator inlet can be calculated according to

$$T_{s1}^z = T_{s1} - z \cdot \Delta T_s^z.$$

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$$T_{s1}^z = T_{s1} - z \cdot \Delta T_s^z.$$

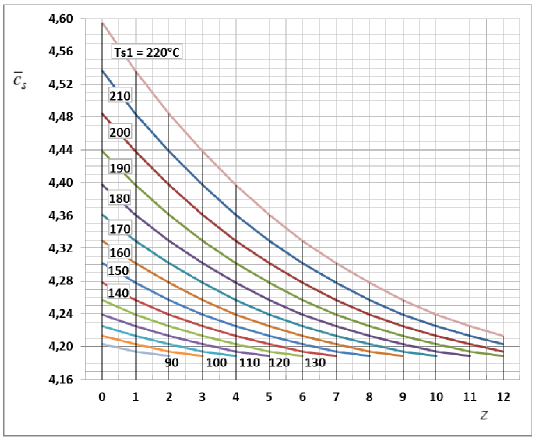

Fig. 4.Average specific heat $\bar{c}_s = f(T_{s1}, z)$

Tab. 1. Average specific heat $\bar{c}_s = f(T_{s1}, z)$

z \ T_{s1}	Specific heat at constant pressure [kJ/(kgK)] for temperatures [°C]:													
	220	210	200	190	180	170	160	150	140	130	120	110	100	90
0	4.595	4.536	4.484	4.438	4.397	4.361	4.329	4.302	4.278	4.257	4.239	4.225	4.213	4.203
1	4.536	4.484	4.438	4.397	4.361	4.329	4.302	4.278	4.257	4.239	4.225	4.213	4.203	4.194
2	4.484	4.438	4.397	4.361	4.329	4.302	4.278	4.257	4.239	4.225	4.213	4.203	4.194	4.189
3	4.438	4.397	4.361	4.329	4.302	4.278	4.257	4.239	4.225	4.213	4.203	4.194	4.189	
4	4.397	4.361	4.329	4.302	4.278	4.257	4.239	4.225	4.213	4.203	4.194	4.189		
5	4.361	4.329	4.302	4.278	4.257	4.239	4.225	4.213	4.203	4.194	4.189			
6	4.329	4.302	4.278	4.257	4.239	4.225	4.213	4.203	4.194	4.189				
7	4.302	4.278	4.257	4.239	4.225	4.213	4.203	4.194	4.189					
8	4.278	4.257	4.239	4.225	4.213	4.203	4.194	4.189						
9	4.257	4.239	4.225	4.213	4.203	4.194	4.189							
10	4.239	4.225	4.213	4.203	4.194	4.189								
11	4.225	4.213	4.203	4.194	4.189									
12	4.213	4.203	4.194	4.189										

Tab. 2. Average specific heat $\bar{c}_{s1} = f(T_{s1}, z)$

z \ T_{s1}	Specific heat at constant pressure [kJ/(kgK)] for temperatures [°C]:													
	220	210	200	190	180	170	160	150	140	130	120	110	100	90
1	4.595	4.536	4.484	4.438	4.397	4.361	4.329	4.302	4.278	4.257	4.239	4.225	4.213	4.203
2	4.566	4.510	4.461	4.418	4.379	4.345	4.316	4.290	4.268	4.248	4.232	4.219	4.208	4.199
3	4.538	4.486	4.440	4.399	4.362	4.331	4.303	4.279	4.258	4.240	4.226	4.214	4.203	4.195
4	4.513	4.464	4.420	4.381	4.347	4.318	4.292	4.269	4.250	4.234	4.220	4.209	4.200	4.193
5	4.490	4.443	4.402	4.365	4.333	4.305	4.281	4.260	4.242	4.227	4.215	4.205	4.197	
6	4.469	4.424	4.385	4.351	4.321	4.294	4.272	4.252	4.236	4.222	4.211	4.201		
7	4.449	4.407	4.370	4.337	4.309	4.284	4.263	4.245	4.230	4.217	4.207			
8	4.430	4.391	4.356	4.325	4.299	4.276	4.256	4.239	4.225	4.213				
9	4.413	4.376	4.343	4.314	4.289	4.267	4.249	4.233	4.220					
10	4.398	4.362	4.331	4.304	4.280	4.260	4.243	4.228						
11	4.383	4.350	4.320	4.295	4.273	4.254	4.238							
12	4.370	4.338	4.311	4.286	4.266	4.248								

Tab. 3.Average specific heat $\bar{c}_{s2} = f(T_{s1}, z)$

z \ T_{s1}	Specific heat at constant pressure [kJ/(kgK)] for temperatures [°C]:													
	220	210	200	190	180	170	160	150	140	130	120	110	100	90
0	4.595	4.536	4.484	4.438	4.397	4.361	4.329	4.302	4.278	4.257	4.239	4.225	4.213	4.203
1	4.566	4.510	4.461	4.418	4.379	4.345	4.316	4.290	4.268	4.248	4.232	4.219	4.208	4.199
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3	4.513	4.464	4.420	4.381	4.347	4.318	4.292	4.269	4.250	4.234	4.220	4.209	4.200	4.193
4	4.490	4.443	4.402	4.365	4.333	4.305	4.281	4.260	4.242	4.227	4.215	4.205	4.197	
5	4.469	4.424	4.385	4.351	4.321	4.294	4.272	4.252	4.236	4.222	4.211	4.201		
6	4.449	4.407	4.370	4.337	4.309	4.284	4.263	4.245	4.230	4.217	4.207			
7	4.430	4.391	4.356	4.325	4.299	4.276	4.256	4.239	4.225	4.213				
8	4.413	4.376	4.343	4.314	4.289	4.267	4.249	4.233	4.220					
9	4.398	4.362	4.331	4.304	4.280	4.260	4.243	4.228						
10	4.383	4.350	4.320	4.295	4.273	4.254	4.238							
11	4.370	4.338	4.311	4.286	4.266	4.248								
12	4.358	4.328	4.302	4.279	4.259									

Tab. 4.Ratios of average specific heat $\bar{c}_s/\bar{c}_{s1} = f(T_{s1}, z)$

T_{s1} / z	Specific heat at constant pressure [kJ/(kgK)] for temperatures [°C]:						
	220	210	200	190	180	170	160
1	0.987160	0.988536	0.989741	0.990762	0.991813	0.992662	0.993763
2	0.982149	0.984035	0.985653	0.987210	0.988582	0.990104	0.991310
3	0.977892	0.980160	0.982281	0.984162	0.986169	0.987839	0.989310
4	0.974243	0.976981	0.979412	0.981912	0.984070	0.985987	0.987767
5	0.971269	0.974298	0.977327	0.979979	0.982370	0.984578	0.986919
6	0.968781	0.972387	0.975562	0.978433	0.981099	0.983855	0.986266
7	0.967052	0.970791	0.974174	0.977307	0.980506	0.983328	0.985859
8	0.965634	0.969566	0.973196	0.976850	0.980109	0.983043	0.985490
9	0.964577	0.968742	0.972880	0.976588	0.979949	0.982790	0.985905
10	0.963913	0.968570	0.972755	0.976556	0.979815	0.983310	
11	0.963892	0.968587	0.972855	0.976546	0.980446		
12	0.964055	0.968824	0.972973	0.977292			

T_{s1} / z	Specific heat at constant pressure [kJ/(kgK)] for temperatures [°C]:						
	150	140	130	120	110	100	90
1	0.994421	0.995091	0.995772	0.996697	0.997160	0.997626	0.997859
2	0.992308	0.993322	0.994586	0.995510	0.996208	0.996673	0.997737
3	0.990652	0.992250	0.993554	0.994636	0.995333	0.996590	
4	0.989693	0.991352	0.992796	0.993839	0.995307		
5	0.988921	0.990713	0.992099	0.993879			
6	0.988399	0.990124	0.992223				
7	0.987919	0.990341					
8	0.988234						

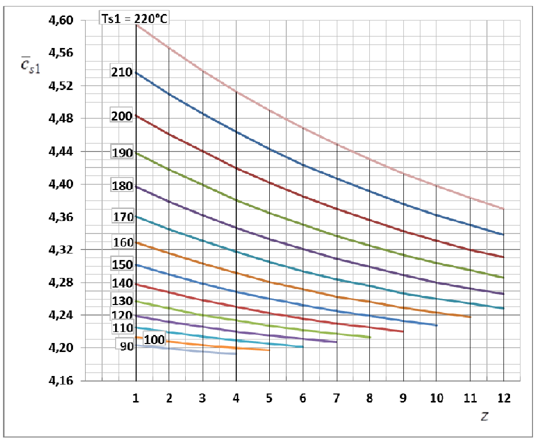

Fig. 5. Average specific heat $\bar{c}_{s1} = f(T_{s1}, z)$

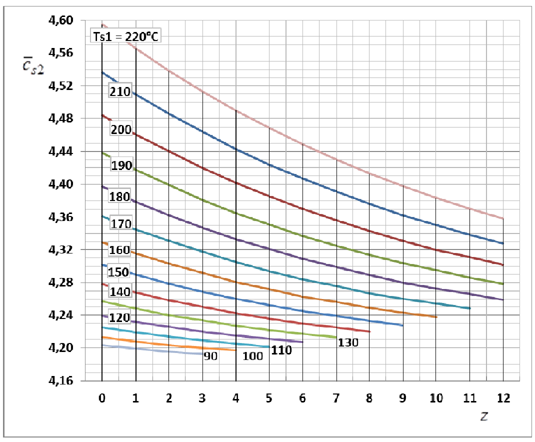

Fig. 6. Average specific heat $\bar{c}_{s2} = f(T_{s1}, z)$

Tab. 5.Ratios of average specific heat $\bar{c}_s/\bar{c}_{s2} = f(T_{s1}, z)$

z \ T_{s1}	Specific heat at constant pressure [kJ/(kgK)] for temperatures [°C]:						
	220	210	200	190	180	170	160
1	0.993538	0.994235	0.994844	0.995359	0.995889	0.996318	0.996872
2	0.988028	0.989300	0.990390	0.991437	0.992359	0.993381	0.994190
3	0.983327	0.985046	0.986652	0.988074	0.989591	0.990851	0.991961
4	0.979287	0.981500	0.983461	0.985477	0.987216	0.988758	0.990189
5	0.975943	0.978489	0.981035	0.983260	0.985265	0.987115	0.989075
6	0.973121	0.976238	0.978979	0.981457	0.983755	0.986129	0.988205
7	0.971051	0.974349	0.977329	0.980087	0.982901	0.985382	0.987605
8	0.969335	0.972856	0.976103	0.979369	0.982280	0.984899	0.987082
9	0.968006	0.971780	0.975525	0.978880	0.981918	0.984484	0.987296
10	0.967086	0.971346	0.975170	0.978642	0.981616	0.984804	
11	0.966801	0.971129	0.975061	0.978459	0.982046		
12	0.966728	0.971153	0.975000	0.979002			
z \ T_{s1}	Specific heat at constant pressure [kJ/(kgK)] for temperatures [°C]:						
	150	140	130	120	110	100	90
1	0.997203	0.997540	0.997881	0.998346	0.998578	0.998812	0.998928
2	0.994859	0.995538	0.996384	0.997002	0.997469	0.997780	0.998490
3	0.992973	0.994176	0.995158	0.995972	0.996495	0.997440	
4	0.991737	0.993070	0.994228	0.995065	0.996242		
5	0.990750	0.992249	0.993407	0.994894			
6	0.990039	0.991523	0.993327				
7	0.989413	0.991538					
8	0.989528						

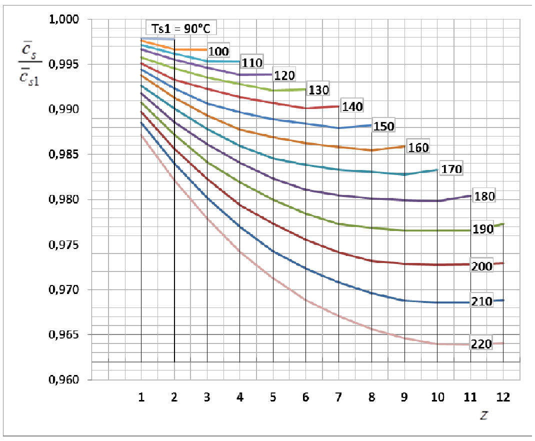

Fig. 7. Ratio of average specific heat $\bar{c}_s/\bar{c}_{s1} = f(T_{s1}, z)$

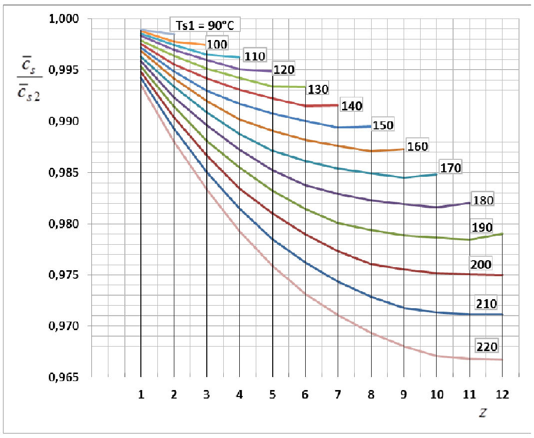

Fig. 8. Ratio of average specific heat $\bar{c}_s/\bar{c}_{s2} = f(T_{s1}, z)$

Tab. 6. Coefficient $\beta = f(T_{s1}, z)$

z \ T_{s1}	Specific heat at constant pressure [kJ/(kgK)] for temperatures [°C]:						
	220	210	200	190	180	170	160
1	0.999917	0.999934	0.999947	0.999957	0.999966	0.999973	0.999980
2	0.999786	0.999829	0.999862	0.999890	0.999913	0.999934	0.999950
3	0.999631	0.999703	0.999763	0.999811	0.999856	0.999889	0.999914
4	0.999466	0.999574	0.999659	0.999737	0.999796	0.999842	0.999880
5	0.999309	0.999447	0.999570	0.999665	0.999740	0.999801	0.999857
6	0.999161	0.999344	0.999486	0.999600	0.999693	0.999776	0.999838
7	0.999046	0.999251	0.999414	0.999548	0.999667	0.999756	0.999825
8	0.998946	0.999174	0.999359	0.999522	0.999648	0.999744	0.999813
9	0.998867	0.999118	0.999336	0.999506	0.999637	0.999733	0.999821
10	0.998812	0.999099	0.999323	0.999499	0.999629	0.999746	
11	0.998801	0.999093	0.999323	0.999495	0.999649		
12	0.998804	0.999101	0.999324	0.999523			
z \ T_{s1}	Specific heat at constant pressure [kJ/(kgK)] for temperatures [°C]:						
	150	140	130	120	110	100	90
1	0.999984	0.999988	0.999991	0.999995	0.999996	0.999997	0.999998
2	0.999960	0.999970	0.999980	0.999987	0.999990	0.999993	0.999997
3	0.999934	0.999955	0.999969	0.999978	0.999984	0.999991	
4	0.999915	0.999940	0.999958	0.999970	0.999982		
5	0.999898	0.999928	0.999948	0.999969			
6	0.999884	0.999916	0.999948				
7	0.999872	0.999918					
8	0.999877						

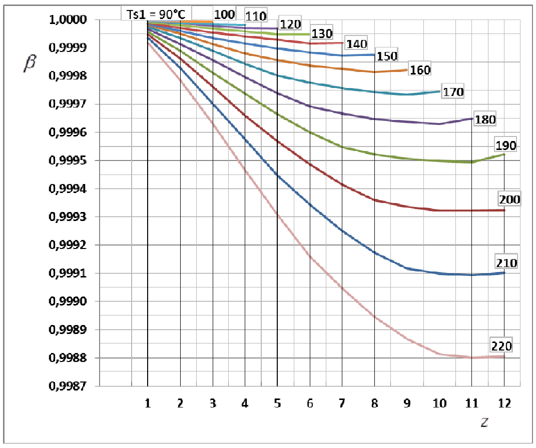

Fig. 9. Coefficient $\beta = f(T_{s1}, z)$

And basing on formula (9a), the temperature of recirculating water at the evaporator outlet can be calculated according to

$$T_{s2}^{z} = T_{s1} - (1+z)\cdot \Delta T_{s}^{z}.$$

The heat flux delivered to evaporating medium, taking into account (10) and

$$\beta = f(T_{s1},\, z) \cong 1\,,$$

takes the following form:

$$\dot{Q}_s = (1+z)\dot{m}_s \bar{c}_s \left(T_{s1}^{z} - T_{s2}^{z}\right) = (1+z)\dot{m}_s \bar{c}_s \Delta T_s^{z}. \quad (10)$$

4. CONCLUSIONS

Utilisation of an evaporator with enhanced flow rate of heat carrier, achieved with heat carrier recirculation leads to lower temperature of heat carrier at evaporator inlet due to mixing of streams with different temperatures. The temperature at evaporator outlet, due to heat transfer, is also lower. However, the mass flow rate of heat carrier increases to $\dot{m}_s(1+z)$. The enhancement of flow rate can be measured with recirculation coefficient z. An ORC power plant with enhanced flow through evaporator, with proper selection of operational characteristics and working fluid, can reach near critical conditions. It also enhances the flow rate of working fluid due to lower latent heat of vaporization at higher temperatures (and reaches 0 at critical point).

The analysis of $\bar{c}_s$, $\bar{c}_{s1}$ and $\bar{c}_{s2}$ shows that the average specific heats do not have significant impact on T_{s1}^z, T_{s2}^z and ΔT_s^z.

The development of computational model and example of its use are presented in the research project [12]. This computational model was used in [13].

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Biographical notes

Tomasz Kujawa received his M.Sc. degree in Mechanical Engineering (specialization: Thermal Energy Systems) from Mechanical Faculty and next Ph.D (with honors) degree in Construction from Faculty of Civil Engineering and Architecture, Technical University of Szczecin, in 1993 and 2003, respectively. Since 1993 he has been a researcher in Department of Heat Engineering, Faculty of Mechanical Engineering and Mechatronics, West Pomeranian University of Technology, Szczecin (earlier Technical University of Szczecin). He currently works as a lecturer. His scientific interests focus on problems of obtaining geothermal energy (ground heat exchangers) and ORC installations. He has participated in over 8 national research projects, presenting results of his work at international and national conferences, published more than 80 scientific papers in international and national journals, book chapters, as well as conference proceedings. He is also the co-author of 2 monographs.

Władysław Nowak received his M.Sc. degree in specialization of Shipbuilding and next Ph.D as well as D.Sc. degree from Gdańsk University of Technology, in 1959, 1965 and 1971, respectively. From 1978 he was as an associated professor, from 1991 to the position of full professor. From 1957 until his retirement, he worked continuously at the Faculty of Mechanical Engineering at the Szczecin University of Technology. In his professional life he performed many functions, among others vice-dean and dean of the Faculty of Mechanical Engineering and rector of the Szczecin University of Technology. In the years 1978-2003 he was the head of the Department of Heat Engineering. Scientific and research interests include: renewable energy sources (with ORC), geothermal heat plants, heat management, ventilation and air conditioning, heat exchange and heat exchangers. His specialty is thermal technology, especially heat transfer, has a significant scientific output, including authorship or co-authorship of 5 monographs, 6 didactic scripts, 5 patents, 3 patent applications and over 450 original scientific publications in major national and international scientific journals and conference materials. He promoted 11 doctors, 4 of whom received a postdoctoral degree and 3 are titular professors. He is a reviewer of 17 habilitation dissertations and 27 doctoral dissertations from Poland and abroad. He made 10 opinions for awarding the title of professor and a dozen for the position of full professor. Promoter of over 250 diploma theses. Organizer and co-organizer of national and international symposia and conferences, including: Heat Transfer and Renewable Sources of Energy. From January 1, 2003, Professor Władysław Nowak is retired, still working actively in the field of science and playing an important role in the activity and development of the parent unit, which is the Department of Heat Engineering.