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Comparative Efficiency of Geothermal Vapor-Turbine Cycles

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ABSTRACT

Optimal design of the geothermal power generation units is considered as a part of development of the Verhne-Mutnovsky Geothermal Power Station. A well drill provides a hot water with temperature of 420 K (162 C) at 0.65 MPa and steam fraction of 0.3. In addition, the IV unit of the station consumes a hot water streams separated from the others three units built earlier. Thermodynamic exergy analysis displays losses due to constraints associated to heat exchangers scaling. The analysis also helps to optimize a bottom cycle configuration, designed to transform geothermal heat to electricity for both summer and winter seasons. The analysis takes into account both the bottom cycle configuration and working fluid properties. Selecting of a working fluid is based on the engineering aspects and environmental characteristics and safety considerations.

Thermodynamic analysis shows influence of both intrinsic and extrinsic losses on the power unit efficiency, which can be varied with cost considerations. Properties of the working fluid in the bottom cycle influence on optimal equipment design: turbine efficiency and size, volumetric flow rate at the turbine outlet, and overall heat transfer coefficient in the heat exchangers are important. It is shown that application of the mixed working fluids provides more possibilities for improving the efficiency. This is due to better temperature profiles in the heat exchangers and pressure ratio in the bottom cycle. In addition, mixed working fluids allow more flexibility for optimal regulation at variable loads and ambient temperature in the field.

1. INTRODUCTION

In the geothermal technology electrical power can be produced from potential sources of different types. Geothermal water-steam mixture from the well drill can be separated at high pressure. Saturated steam can be directly used in a steam turbine for power production. Different source of power – is a heat flux at a relatively high temperature. It can be obtained from hot water separated from the pressurized saturated steam. Another heat source is condensing low-pressure steam from the turbine outlet. The heat source temperature varies in a wide range from maximal corresponding to saturated vapor steam from the well drill to minimal, which may be close to the ambient temperature (T_A).

To generate electricity from waste heat in a given temperature range a bottom cycle can be used.

Rozenfeld,L. and Tkachev,A. (1955) analyzed efficiency of the cycle. The world-first geothermal power station operated with Freon R-12 was built in 1965-1967 in the USSR at Kamchatka Region (Moskvichova A.L., Popov A.Ye., 1970). The station was a basic part of the Paratunskiy Power Plant. Since that, different bottom cycles and working fluids (WF) have been considered. A traditional Rankine cycle, and more complex in design Kalina cycle have been analyzed for practical applications (Kalina A.I., 1984). A literature review and analysis show (Boiarski M.Y., 2003) et al. that bottom cycles efficiency depends on both the cycle configuration and working fluid properties, which can be either pure or mixture.

A goal of this paper is a comparison efficiency of the Rankine bottom cycle designed for the IV unit of Verhne-Mutnovsky Geothermal Power Station (Povarov O. et al., 2003). The IV power generation unit consumes both hot water and saturated steam from the well drill at P = 0.65 MPa and T = 435 K (162 C). In addition, the hot water that separated from the well drills feeding the power units III, II, and I built earlier is supplied to the IV unit at temperature of 417 K (144 C). As long as the power station is remotely placed, the units should be designed for a long-term operation with minimal maintenance at both summer and winter seasons. According to this, it is assumed that neither the steam turbine nor the bottom cycle should operate under vacuum.

The unit consists of two circuits. The first one generates power with steam expanding in turbine (T1) from P_H = 0.65 MPa to P_L = 0.11 MPa. Saturated water steam is directed to the condenser-evaporator and condenser-heater that both are in a heat transfer relationship with the second circuit working fluid WF. This allows cooling the water condensate down to the ambient temperature before returning to the injection well. Hermetically sealed, closed loop the second circuit presents the bottom cycle operating the Rankine cycle (RC) with recuperation. The RC circuit configuration provides for WF heating, evaporating and vapor superheat from different sources in a wide temperature range. This allows to transforming high temperature heat into electrical power in T2 the turbine. Both, turbine (T2) and feed pump operate at pressure ratio PR = P_H/P_L, which depends of a selected working fluid in the RC circuit. An air-cooled condenser is used in the bottom cycle to remove the heat to the environment. The second circuit can be simplified when operating without the recuperative heat exchanger. This is trade off between reducing the equipment cost and bottom cycle efficiency. Appropriate information on the decision is obtained below from computer modeling.

Two configurations of the Rankine cycle are considered at this analysis: with and without heat recuperation. In the first case a counter flow heat exchanger must be used as it is shown in Figure 1, which presents a simplified schematic of the IV power unit.

2. MODELING OF THE CYCLE EFFICIENCY AND WORKING FLUID PROPERTIES

Exergy analysis (Brodyansky V. et al., 1994) can be applied to compare different approaches in terms of work power that can be obtained with an ideal, fully reversible cycle. An exergy (E) of a given stream at specified parameters P and T can be calculated as specific thermodynamic properties enthalpy (h) and entropy (s) and flow rate (G) are known:

$$E(P,T) = Ge(P,T) = G\{[h(P,T) - h(P_A,T_A)] - T_A[s(P,T) - s(P_A,T_A)]\} \quad (1)$$

At this analysis thermodynamic properties of working fluids have been modeling with Peng-Robinson equation of state (Reid, R.C. et al., 1987) for both the bottom cycle working fluids and water-steam flows. This simple cubic equation of state allows quick calculations with reasonable accuracy. By estimation, discrepancy in calculations of a mass flow rate (G) from the heat exchanger balance would not increase 5…10 % compared to a standard data on thermodynamic properties. This is acceptable for a comparative analysis.

Constituents of the supplied exergy brought to (E_{BT}) the IV unit are presented below in Table 1. Not all the E_{BT} could be used in the bottom cycle. The separated water temperature from the well drill should not be below 393 K (120 C) to avoid potential scaling in the heat exchangers. This technological constrain causes the exergy loss D_{CN}. Only reduced - net exergy $E_{NT} = E_{BT}$-D_{CN} can be used for power production. Meanwhile, the condensate from the steam turbine can be cooled down to T_A without constrains. One of the advantages of the bottom cycle is associated with possibility to generate electricity at any ambient temperature (T_A). Table 1a shows data for a summer season and Table 1b – for the winter season. It is assumed that sink temperature is essentially close to the ambient one. Each constituents in Table 1 is calculated as $E = Ge(P,T)$.

Modeling shows:

- The bottom cycle could provide an essential increase in the IV unit power production as it consumes about 50 % of the exergy E_{BT} supplied to the unit. Thus, the bottom cycle efficiency is important providing a high overall performance of the IV Unit.

- Exergy available for transformation in the steam turbine presents a relatively small part in overall balance: 20 % E_{BT} for summer, and 15% E_{BT} for winter season.

- In winter season, the bottom cycle could consume about 25% as higher exergy due to lower ambient temperature.

- Constraint on the separated water temperature T = 393 K (120 C) due to heat exchanger scaling causes of 30 to 35 % loss of the exergy brough to the power unit from the well drill.

Data presented in Table 1a and 1b can be used to make assessments on the thermodynamic efficiency of the bottom Rankine cycle (RC). In terms of the exergy analysis the RC efficiency (EEF_{RC}) can be determine as following:

$$EEF_{RC} = \frac{PW_{T2}}{E_{NT}} = 1 - \frac{D_{SUM}}{E_{NT}} = 1 - \frac{(D_{INT} + D_{EXT})}{E_{NT}} \quad (2)$$

In this equation power production (PW_{RC}) in the Rankine cycle is compared to net exergy E_{NT} supplied to the RC for transformation. Summarized losses D_{SUM} includes of two constituents. D_{INT}- intrinsic inherently belongs to both the cycle configuration and thermodynamic properties of the selected working fluid. D_{EXT} – presents the losses depending of the equipment performance. The value of this loss varies according to the quality of design and the equipment cost.

Overall efficiency of the units EFF_{BT} can be determined in a similar manner.

$$EEF_{BT} = \frac{PW_{T1} + PW_{T2}}{E_{BT}} = 1 - \frac{(D_{INT} + D_{EXT} + D_{CN})}{E_{BT}} \quad (3)$$

Cycle modeling allows evaluating of power generated and efficiency either for the unit or particular part of it with some assumptions that take into account factors causing the exergy losses. Some results are presented in the next section for both summer and winter seasons.

3. MODELING OF THE UNIT IV EFFICIENCY

Results of previous research shows that the bottom cycle efficiency depends on both the RC configuration and working fluid properties (Boiarski M.Y. et al., 2003). Two bottom cycle configurations have been considered: without and with recuperation. Figure 1 presents a configuration in general case, which includes recuperative heat exchangers.

The IV unit can operate at different regimes. In a minimal heat load regime, the separate is supplied to the unit from a single well drill, only. In a full load regime, the separate is additionally supplied from other units I, II and III of the same GPS has been operating since 1999**.** The additional heat sources allow WF in the RC superheating for elevated temperatures.

A modern working fluid (WF) in the bottom cycle should match different requirements that identical to those developed for applications industrial refrigerants (ASHRAE HANDBOOK, 1997). For evaluation efficiency of the RC cycle R21 the refrigerant has been selected, as its thermodynamic properties allow efficient operating in both seasons summer and winter. Table 2 shows the WF parameters in the selected points of the RC for summer and winter seasons in the full load regime. Efficiency of both unit IV and RC based on modeling data is given in Table 3 and 4. It is assumed that a sink temperature (T_{SINK}) in the RC equals to the ambient temperature: T_A = 303 K (30 C) – for summer, and T_A = 278 K (+5 C) - for the winter season.

Extrinsic losses significantly influence on the efficiency of the RC and overall unit IV. This assessment has been conducted to evaluate a maximal expected power production with the following assumptions: a minimal temperature difference in the heat exchangers – ΔT_{MIN} = 5 K, adiabatic efficiency of turbines TB_1 – η_S = 0.73 and TB_2 – η_S = 0.83. All the others factors like hydraulic pressure drop, heat losses through the insulation are negligibly small, by the assumption.

Figure 2 presents temperature – heat load diagrams for the RC heat exchangers at ΔT_{MIN} = 5 K. Influencing of ΔT_{MIN} on the power efficiencies is shown below in figure 3.

The modeling analysis shows:

- The bottom cycle efficiency is in a range of EFF_{RC} = 0.65 to 0.70. During winter season the bottom cycle allows of 20 % in power production increase compared to summer season.

- Power of the steam turbine TB_1 is identical for both seasons PW_{TB1} = 2070 kW. This is due to constrain for the turbine pressure outlet P_L = 0. 11 MPa.

- Heat recuperation in the bottom cycle improves the efficiency up to 5 to 7 %. This provides almost 300 kW increase in power production for the full-load regime.

- Efficiency EFF_{BT} = 0.6 is significantly less then EFF_{RC} = 0.70…0.72 due to constrain on the T_{SP} = 120 C.

Maximal efficiency of the RC obtained from modeling is up to EEF_{RC} =0.70. Nearly 25 % of the lost exergy is due to temperature profiles in the heat exchangers (HX). This can be seen from figure 2 calculated at ΔT_{MIN} = 5 K, which is probably the lowest practical number achievable in modern HX. Two factors cause these losses. The first one is because of indirect influence of the constraint on separate temperature T > 120 ^oC. However, this part is relatively small - less than 10%. The second part is caused by the thermodynamic properties of the WF: for instance, D_{EX} = 60% in the recuperative heat exchanger.

The analysis also shows influencing the WF properties on the optimal equipment design. Table 5 presents data for R21. Both pressure ratio P_H /P_L and the WF flow rate – G_{WF} are important in designing the turbine BT_2 to optimize efficiency and size. A production KF of the overall heat transfer coefficient K and heat transfer area F influence on the summarized HX size in the RC. We can see that R21 could be used for both seasons. However, the sink temperature T_{SINK} should not be less than 288 K, to avoid vacuum in the RC. Figure 3 shows influencing of ΔT_{MIN} on the overall heat exchanger size $KF_\Sigma = f(\Delta T_{MIN})$ and power production $PW_{RC} = f(\Delta T_{MIN})$ for both seasons.

Operating pressure P_H and P_L in the bottom cycle are presented for the selected points. Parameter $KF_\Sigma = \Sigma KF_I$ is associated to the HX cost. For each HX is calculated as $KF_I = Q_I / \Delta T_{AV}$, where ΔT_{AV} is average mean temperature. If it is assumed that overall heat transfer coefficient is identical, then an overall HX size is proportional to KF_Σ. Data from Table 5 shows that the recuperative bottom cycle needs only nearly 15 % of increased heat exchangers compared to RC without recuperation. Increasing of ΔT_{MIN} from 5 K to 15 K would reduce the overall HX size in 2.5 times. However, a potential HX cost reduction is connected to reducing in power production nearly 1.5 times. Selecting of an appropriate ΔT_{MIN} is trade off between long term and initial investments.

4. WORKING FLUID INFLUENCE ON THE BOTTOM CYCLE EFFICIENCY

Different working fluids have been considered in the Rankine Cycle developing (Boiarski M.Y. et al., 2003). Modern international regulations require providing not only safe operation in the field but acceptable environmental parameters (ASHRAE HANDBOOK, 1997). Some novel environmentally friendly high boiling refrigerants can be efficiently used as in the bottom cycle. Evaluations (Boiarski M.Y. et al, 2003; Zyhowski, G., 2003) show that R245fa would be acceptable in some applications. Meanwhile, R245fa has some drawbacks. One of them is associated with a relatively low pressure in the condenser, especially at low ambient temperature in the winter season. Potential toxicity of R245fa may be among the other issues. Developing of highly efficient WF with appropriate P_H and P_L in the RC is still the issue.

Table 6 demonstrates the WF influence on the efficiency and the design parameters. Refrigerants R21, R142b have been selected at this step. The comparative analysis data is presented for both summer and winter seasons for the recuperative RC. Efficiency data are included in the table: EFF_{NT} - for RC, EFF_{BT} for - overall unit IV, and traditional thermal coefficient of performance $COP_T = PW_{UNIT} / Q_{SUM}$ for the unit IV. PW_{UNIT} consists of both power of steam turbine T1 and T2 of the RC. Table also presents design parameters for different working fluids. The heat exchangers design parameter KF_Σ calculated at ΔT_{MIN} = 5 K for each heat exchanger.

Cycle modeling combined with the exergy analysis allows explicitly see the WF influence on the performance and design parameters. For this purpose the highly idealized cycle performance was calculated taking into account the intrinsic exergy losses only. At this step it was assumed that ΔT_{MIN} in each HX is negligibly small. By definition, at this limited case the parameter KF_Σ is indefinitely large. Modeling predicts for both WF an identical performance data and KF_Σ. However, the WF flow rate is essentially different, which is important for the station equipment design.

Applications of mixed working fluids (MWF) based on industrially available refrigerants may help in further improvements of the efficiency and design parameters. Mixed refrigerant technology also allows compromising design parameters and environmental characteristics. Table 7 presents data obtained elsewhere (Boiarski M.Y. et al, 2003) on the expecting properties of new MWF operating in the RC compared to R-21 and R-245fa. The table demonstrates a possibility compromising properties of pure WF matching certain requirements within geothermal technology.

Table 7: Data for New MWF for RC operating at T_H = 100 C, T_L = 30 C.

Working Fluid	R-21	R-245fa	MWF-1	MWF-2
P_H/P_L, bar	11/2.2	10.3/1.8	12.1/2.4	16.2/3.0
MM	103	134	114	90
ODP	0.01	0.00	< 0.01	0.00
GWP (100 yr.)	210	950	< 600	< 500

Data presented above obtained for the IV unit of the Verhne-Mutnovsky Geothermal Power Station located in the Kamchatka region also demonstrate different trends that could be of general interest in geothermal technology.

CONCLUSION

1. The bottom Rankine cycle utilizing the heat from the separated water and condensing steam after the turbine is important part of the geothermal station at this application. In the full-load regime its fraction in the station power production is above 50%.

2. In the winter season, the bottom cycle could increase power production about 25 % compared to the summer season.

3. Recuperation in the bottom Rankine cycle provides 5 to 7 % increase in power production and needs about 15 % of increased area in heat exchangers.

4. The bottom cycle efficiency significantly depends on the heat exchangers efficiency: increasing a minimal

temperature difference from 5 K to 15 K would reduce power production by 2.5 times.

5. Developing of working fluid efficiently operating in the bottom cycle without vacuum in summer and winter seasons is important step in the optimal station design. The solution could be found with applications of mixed working fluids based on available industrial refrigerants.

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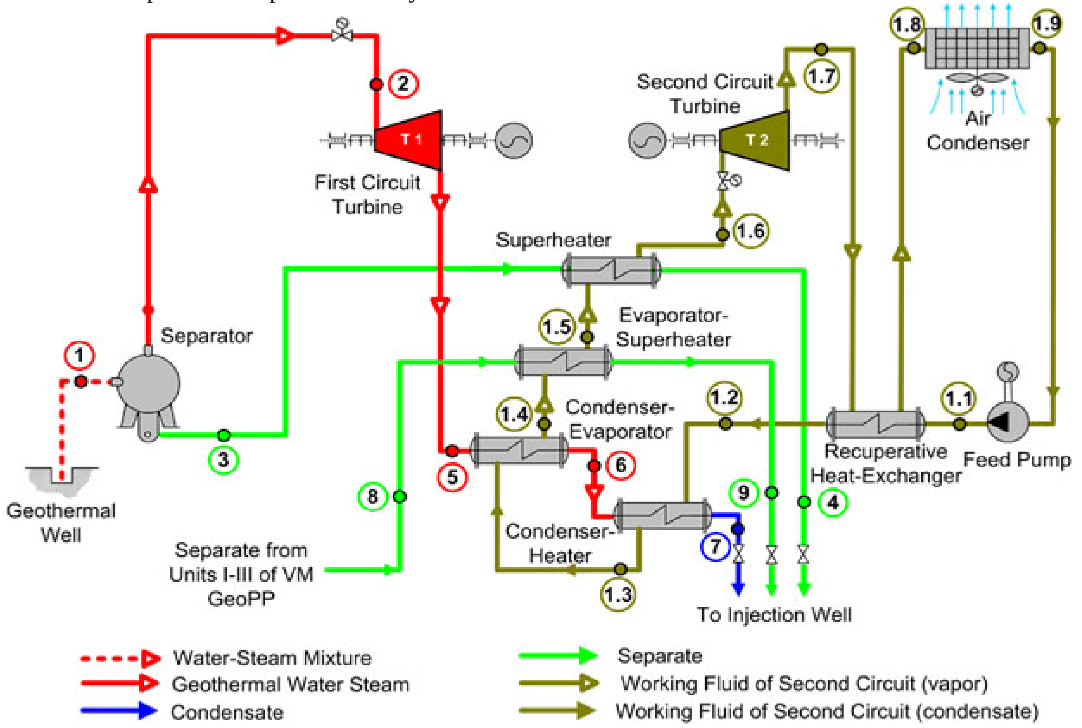

Figure 1: Schematic of the IV Unit.

Table 1a: Summer season T_A = 303 K: Constituents of the exergy supplied to the IV unit.

E_{BT}: Brought To Exergy, kW (% BT)	**Well Drill (IV):** E= 33.9*283.1= **9600**		**Separate from Units I, II, III:**	**Summary, kW (%)**
	Hot Water: E=24.7*106.7 = **2640** (20 %)	*Steam:* E=9.2*758.4= **6960** (51 %)	E=50.5*78.0 = **3940** (29 %)	**13540** (100 %)
D_{CN}: Loss due to constrains, kW	*Separate* T= 393 K: D=24.7*51.8 = **1280**	*Condensate:* T_A, P_L= 0.11 MPa D = 0.1	*Separate:* T= 393 K, D=50.5*51.5=**2600**	**3880** (29 % BT)
E_{NT} = E_{BT} - D_{CN}: Net Exergy to Use in the RC, kW (%)	**1360** (20 % E_{NT})	*Condensing steam* **4260** (61 % E_{NT})	**1340** (19 % E_{NT})	**6960** (51%BT) (100 % E_{NT})
E_{T1}, kW (%BT)				**2700** (20 %BT)

Table 1b: Winter season T_A = 278 K: Constituents of the exergy supplied to the IV unit.

E_{BT}: Brought To Exergy, kW (% BT)	**Well Drill (IV):** E= 33.9*362.8= **12300**		**Separate from Units I, II, III:**	**Summary, kW (%)**
	Hot Water: E=24.7*153.0= **3785** (21 %)	*Steam:* E=9.2*927.7= **8515** (46 %)	E=50.5*119.3=**6025** (33 %)	**18325** (100% BT)
D_{CN}: Loss due to constrains, kW	*Separate* T= 393 K: D=24.7*86.2= **2130**	*Condensate*: T_A, P_L= 0.11 MPa D = 0.1	*Separate:* T= 393 K, D=50.5*85.9=**4335**	**6465** (35 % BT)
E_{AV} = E_{BT} - D_{CN}: Net Exergy-Available to Use, kW	**1655** (18 % E_{NT})	*Condensed steam* **5865** (63% E_{NT})	**1690** (19% E_{NT})	**9210** (50%BT) (100 % E_{NT})
E_{T1}, kW (%BT)				**2650** (15 %BT)

Table 2: WF parameters in the selected points for summer and winter seasons.
R-21 (Summer: 121.5 kg/s, Winter:113.4 kg/s).

№	**Season**	**Tempera-ture, ^oC**	**Pressure, MPa**	**Enthalpy, kJ/kg**	**Entropy, kJ/(kg·K)**	**Exergy, kJ/kg**
1.1	Summer	35.6	1.27	-92.9	0.93	19.4
	Winter	10.6		-117.3	0.85	1.1
1.2	Summer	64.0	1.27	-63.6	1.02	21.4
	Winter	35.8		-92.8	0.93	3.3
1.3	Summer	98.7	1.27	-24.0	1.14	24.7
	Winter					13.7
1.4	Summer	98.6	1.27	114.1	1.51	50.6
	Winter	98.7		107.6	1.49	47.9
1.5	Summer	103.2	1.27	160.1	1.63	60.2
	Winter	99.2		157.1	1.62	61.2
1.6	Summer	157.4	1.27	200.9	1.73	70.7
	Winter					61.2
1.7	Summer	86.3	0.25	162.1	1.75	25.9
	Winter	55.7	0.11	144.3	1.77	6.8
1.8	Summer	40.6	0.25	132.8	1.67	20.8
	Winter	15.6	0.11	119.9	1.69	4.6
1.9	Summer	35.0	0.25	-93.8	0.93	18.5
	Winter	10.0	0.11	-118.3	0.85	0.1

Table 3: Full heat-load regime: efficiency of Unit IV operating with R21.
a) Summer season: G_{WF} = 114 kg/s (no-recuperation), G_{WF} = 122 kg/s (with recuperation).

Constituents	**Rankine Cycle - RC**		**Unit 4**	
	No-recuperation	With recuperation	RC no-recuperation	RC with recuperation
Exergy used, kW	E_{NT}=6935	E_{NT}=6840	E_{NT}=9635 E_{BT}=13535	E_{NT}=9540 E_{BT}=13535
Turbine power, kW	TB_2=4400	TB_2=4715	TB_1+TB_2: 6470	TB_1+TB_2: 6785
Water pump, kW	105	115	105	115
Efficiency	**EFF_{RC} = 0.62**	**EFF_{RC} =0.67**	**EFF_{NT} = 0.66 EFF_{BT} = 0.47**	**EFF_{NT} = 0.70 EFF_{BT} = 0.49**

b) Winter season: G_{WF} = 108 kg/s (no-recuperation), G_{WF} = 113 kg/s (with recuperation).

Exergy Constituents	**Rankine Cycle - RC**		**Unit 4**	
	No-recuperation	With recuperation	RC no-recuperation	RC with recuperation
Exergy used, kW	E_{NT}=9200	EX_{NT}=9105	E_{NT} =11850 E_{BT} =18325	E_{NT}=11755 E_{BT}=18325
Turbine power, kW	TB_2 =6105	TB_2 =6415	TB_1+TB_2: 8175	TB_1+TB_2: 8485
Water pump, kW	110	115	110	115
Efficiency	**EFF_{RC} = 0.65**	**EFF_{RC} =0.69**	**EFF_{NT} =0.68 EFF_{BT} = 0.44**	**EFF_{NT} = 0.71 EFF_{BT} = 0.46**

Table 4: A single-well regime: efficiency of Unit IV operating with R21.

a) Summer season: G_{WF} = 95 kg/s (no-recuperation), G_{WF} = 101 kg/s (with recuperation).

Exergy Constituents	**Rankine Cycle - RC**		**Unit 4**	
	No-recuperation	With recuperation	RC no-recuperation	RC with recuperation
Exergy used, kW	E_{NT}=5600	E_{NT}=5530	E_{NT}=8300 E_{BT}=9600	E_{NT}=8230 E_{BT}=9600
Turbine power, kW	TB_2=3660	TB_2=3890	TB_1+TB_2: 5730	TB_1+TB_2: 5960
Water pump, kW	88	93	88	93
Efficiency	**EFF_{RC}=0.64**	**EFF_{RC}=0.69**	**EFF_{RC} = 0.68** **EFF_{BT} = 0.59**	**EFF_{RC} = 0.71** **EFF_{BT} = 0.61**

b) Winter season: G_{WF} = 91 kg/s (no-recuperation), G_{WF} = 95 kg/s (with recuperation).

Exergy Constituents	**Rankine Cycle - RC**		**Unit 4**	
	No-recuperation	With recuperation	RC no-recuperation	RC with recuperation
Exergy used, kW	E_{NT}=7515	E_{NT}=7435	E_{NT}=10165 E_{BT}=12300	E_{NT}=10080 E_{BT}=12300
Turbine power, kW	TB_2=5110	TB_2=5330	TB_1+TB_2: 7180	TB_1+TB_2: 7405
Water pump, kW	90	95	90	95
Efficiency	**EFF_{RC}= 0.67**	**EFF_{RC}= 0.70**	**EFF_{RC} = 0.70** **EFF_{BT} = 0.58**	**EFF_{RC} = 0.72** **EFF_{BT} = 0.59**

Table 5: Equipment design parameters.

a) Summer season: T_{SINK} = 303 C

Rankine cycle Parameters	**RC turbine (BT_2): P_H = 1.27 MPa, P_L = 0.25 MPa**			
	Full Load Regime		**Single-well Regime**	
	No-recuperation	With recuperation	No-recuperation	With recuperation
KF	4210	4810	4115	4525
G_{WF}	114	122	95	101

b) Winter season: T_{SINK} = 278 K

Rankine cycle Parameters	**RC turbine (BT_2): P_H = 1.27 MPa, P_L = 0.11 MPa**			
	Full Load Regime		**Single-well Regime**	
	No-recuperation	With recuperation	No-recuperation	With recuperation
KF	4040	4485	4005	4335
G_{WF}	108	113	91	95

Table 6: Influence of the WF properties on performance of the unit with recuperative RC.

a) Summer

WF	**ΔT_{MIN}**	**Power Efficiency**			**Design Parameters**		
		PW_{UNIT}	RC: EFF_{NT}	COP_T	KF_Σ	G_{WF}	RC: P_H / P_L
R21	0	8405	0.90	0.24	-	118	1.40 / 0.22
	5	6785	0.67	0.20	4810	122	1.27 / 0.25
R142b	0	8415	0.88	0.24	-	129	2.24 / 0.39
	5	6788	0.66	0.20	4910	134	2.03 / 0.45

b) Winter

WF	**ΔT_{MIN}**	**Power Efficiency**			**Design Parameters**		
		PW_{UNIT}	RC: EFF_{NT}	COP_T	KF_Σ	G_{WF}	RC: P_H / P_L
R21	0	10150	0.87	0.28	-	110	1.40 / 0.10
	5	8485	0.69	0.24	4485	113	1.27 / 0.11
R142b	0	10420	0.89	0.29	-	116	2.24 / 0.17
	5	8510	0.69	0.25	4680	121	2.03 / 0.21

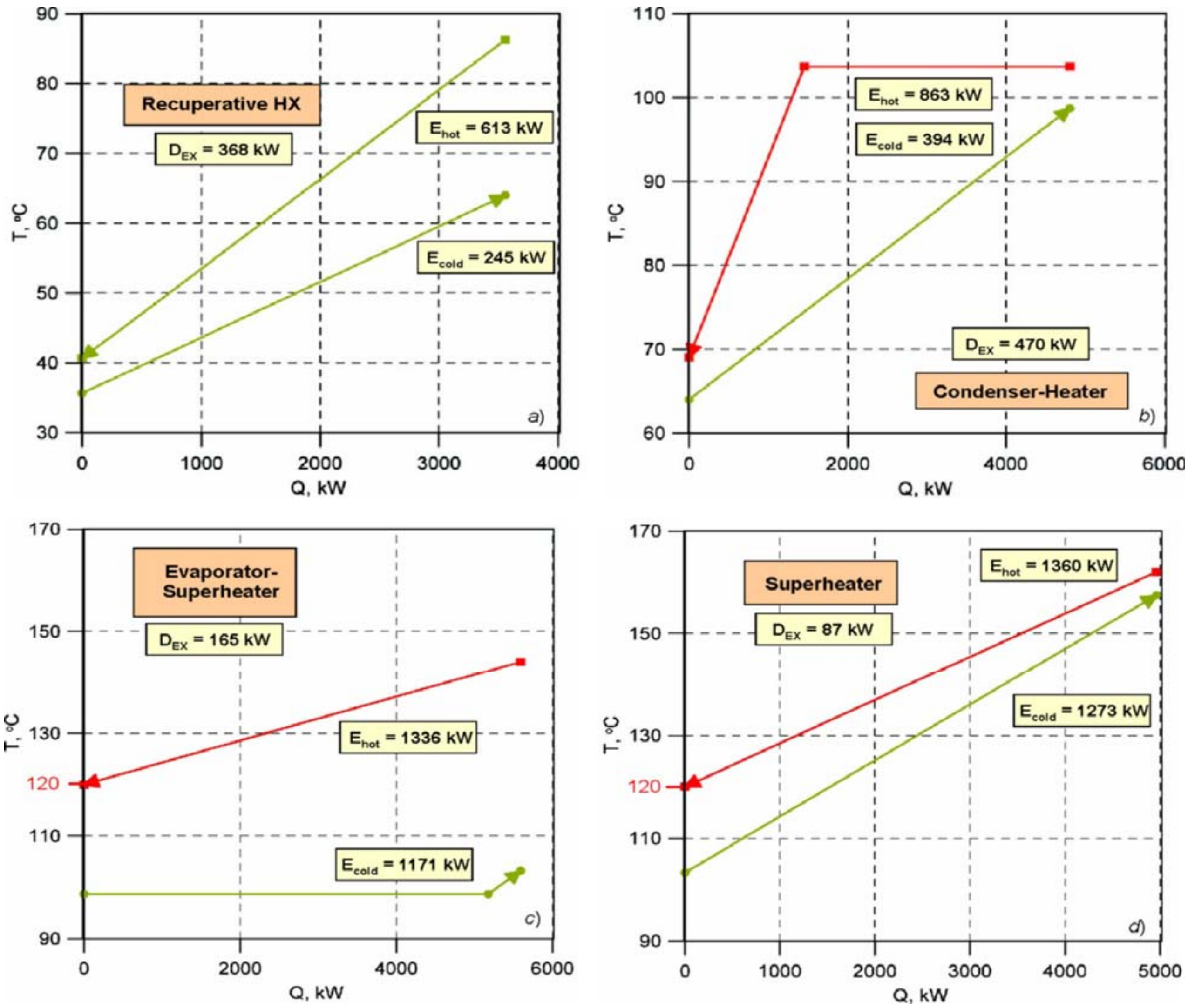

Figure 2: Temperature profiles in the heat exchangers: recuperative cycle - summer season.

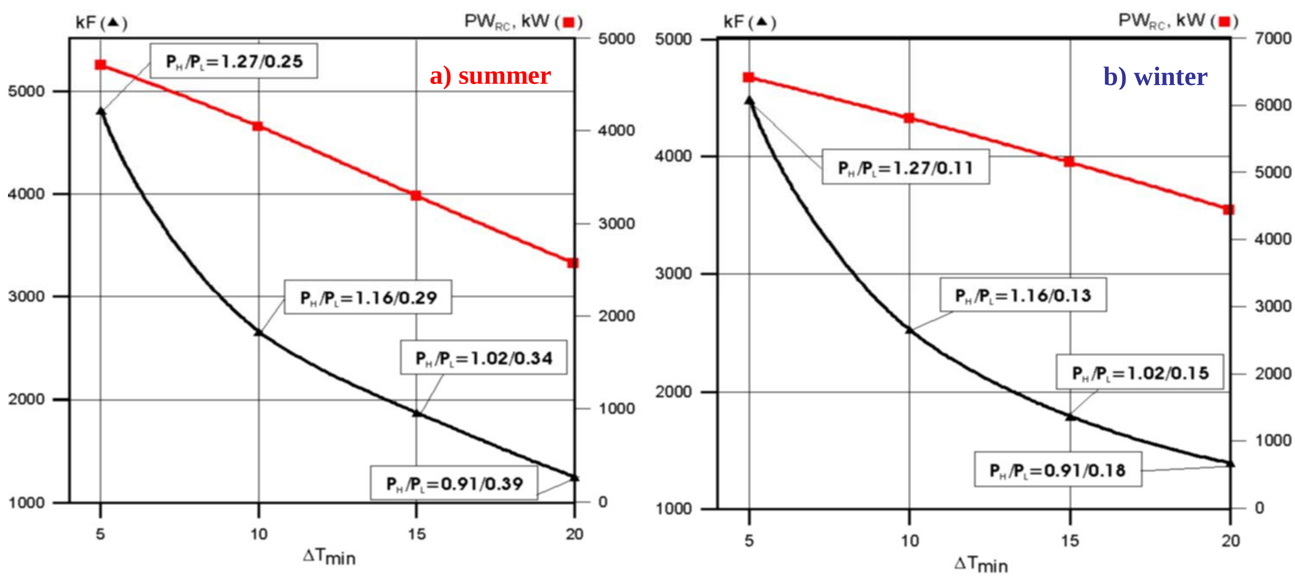

Figure 3: The HX size (KF_Σ) and power production (PW_{RC}) for R21 for Full-load Regime.