In the name of God

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Energy and Exergy Analyses of a New Combined Cycle for Producing Electricity and Desalinated Water Using Geothermal Energy

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- Introduction
- Basic Concept
- Lit. Review
- Model Validation
- Analysis and Result

Waste heat utilization is one of the challenging tasks for researchers

Waste Heat Utilization →
- Reduction fossil fuel
- Reduction environmental problem

Kalina cycle → Waste Heat Utilization → $LiBr/H_2O$ absorption heat transformer

Ammonia-Water Processes

Properties of Ammonia-Water Mixtures

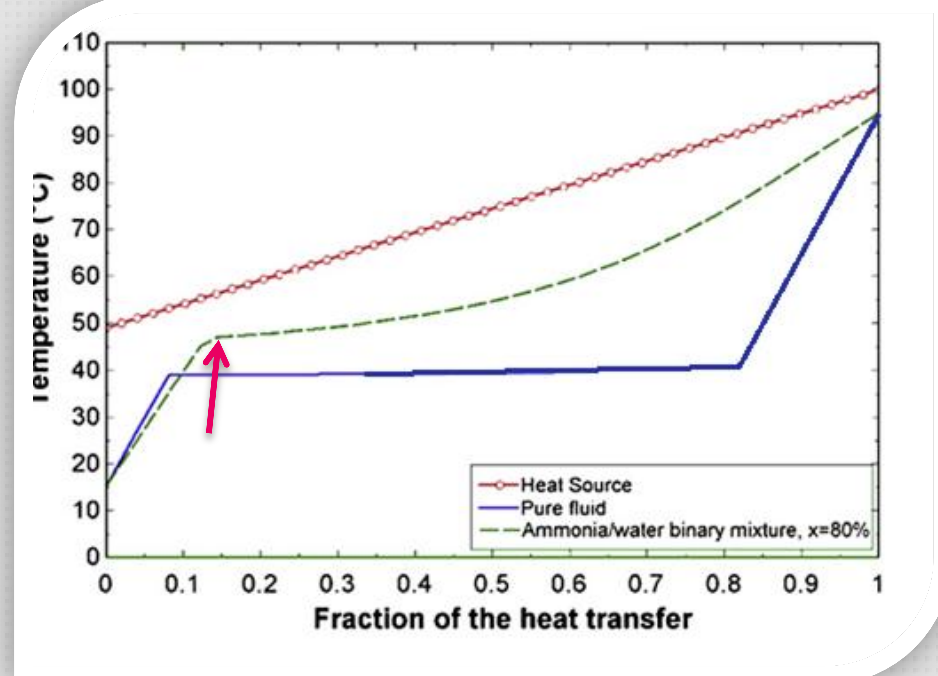
Ammonia and water
- Have different boiling temperature
- Evaporate over large temperature range
- Inexpensive and extensively used in industry
- Have approximately the same molecular weight

Some properties of ammonia and water

	Ammonia (NH_3)	Water (H_2O)
Molecular weight [kg/kmol]	17.0	18.0
Boiling point at 1.013 bar [K]	239.8	373.2
Freezing point at 1.013 bar [K]	195.4	273.2
Critical temperature [K]	405.5	647.3
Critical pressure [bar]	113.5	221.2

Ammonia-Water mixture

- Excellent properties thermo-physical properties
- Environmentally- friendly material
- Best substance for solving global-warming problem
- Non-azeotropic mixture
- Reduce irreversibility loss during heat addition



Previous Work on Ammonia-Water Cycle

- **[Maloney and Robertson, 1950]**
  - First used an ammonia–water mixture
- **[Kalina, 1984]**
  - Employed an ammonia–water mixture as the bottoming cycle working fluid
- **[Park and Sonntag ,1990]**
  - Compared Kalina bottoming cycle for a gas turbine with a single-pressure steam bottoming cycle
- **[Desideri and Bidini ,1996]**
  - Compared steam flash cycles, Rankine cycles with ammonia or ammonia-water mixtures as working fluids and Kalina cycles without separators for geothermal applications

❑ [Goswami, 2000]

- A combined thermal power and refrigeration cycle
- Characteristics of this cycle is that it can use low heat source temperatures bellow 200 C

❑ [Wang et al.’s work, 2003]

- Used binary cycle with mid and low temperature heat recovery

❑ [Liu and Zhang, 2007]

- Cogeneration system of refrigeration and power

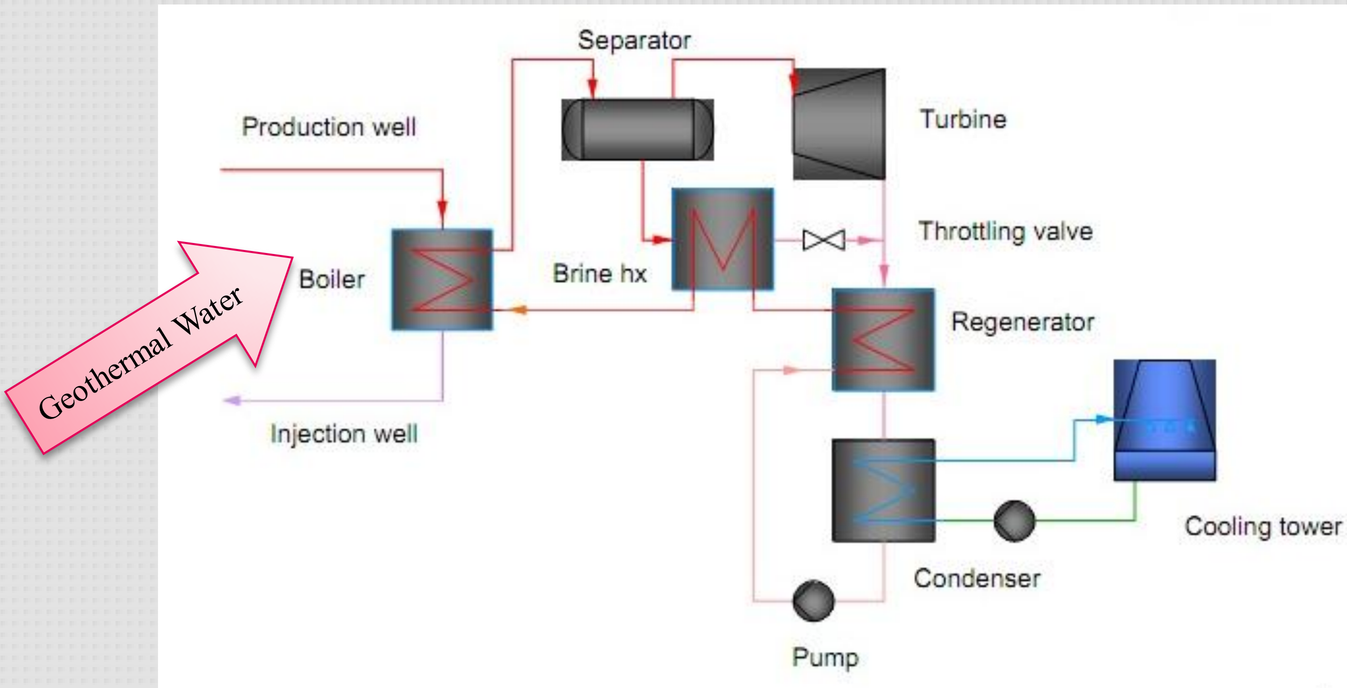
❑ [Sirko Ogriseck, 2009]

- Using geothermal power plant as a heat source of Kalina cycle

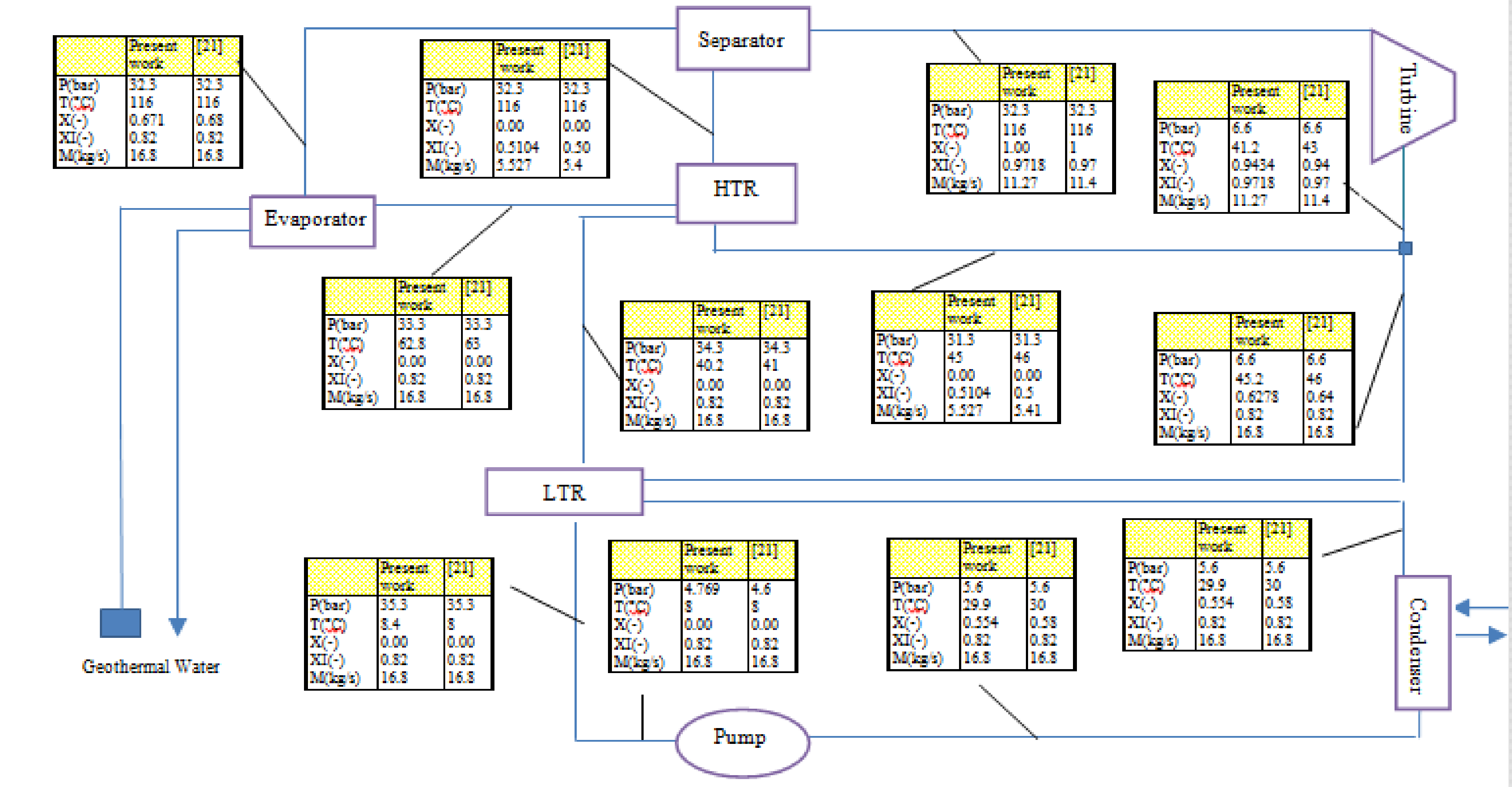
❑ [Philippe Roy, 2010]

- Thermodynamic Analysis and Result of using ammonia-water in the organic Rankine cycle

Kalina Cycle



Model validation



	Present work	[21]
P(bar)	32.3	32.3
T(°C)	116	116
X(-)	0.671	0.68
XI(-)	0.82	0.82
M(kg/s)	16.8	16.8

	Present work	[21]
P(bar)	32.3	32.3
T(°C)	116	116
X(-)	0.00	0.00
XI(-)	0.5104	0.50
M(kg/s)	5.527	5.4

	Present work	[21]
P(bar)	32.3	32.3
T(°C)	116	116
X(-)	1.00	1
XI(-)	0.9718	0.97
M(kg/s)	11.27	11.4

	Present work	[21]
P(bar)	6.6	6.6
T(°C)	41.2	43
X(-)	0.9434	0.94
XI(-)	0.9718	0.97
M(kg/s)	11.27	11.4

	Present work	[21]
P(bar)	33.3	33.3
T(°C)	62.8	63
X(-)	0.00	0.00
XI(-)	0.82	0.82
M(kg/s)	16.8	16.8

	Present work	[21]
P(bar)	34.3	34.3
T(°C)	40.2	41
X(-)	0.00	0.00
XI(-)	0.82	0.82
M(kg/s)	16.8	16.8

	Present work	[21]
P(bar)	31.3	31.3
T(°C)	45	46
X(-)	0.00	0.00
XI(-)	0.5104	0.5
M(kg/s)	5.527	5.41

	Present work	[21]
P(bar)	6.6	6.6
T(°C)	45.2	46
X(-)	0.6278	0.64
XI(-)	0.82	0.82
M(kg/s)	16.8	16.8

	Present work	[21]
P(bar)	35.3	35.3
T(°C)	8.4	8
X(-)	0.00	0.00
XI(-)	0.82	0.82
M(kg/s)	16.8	16.8

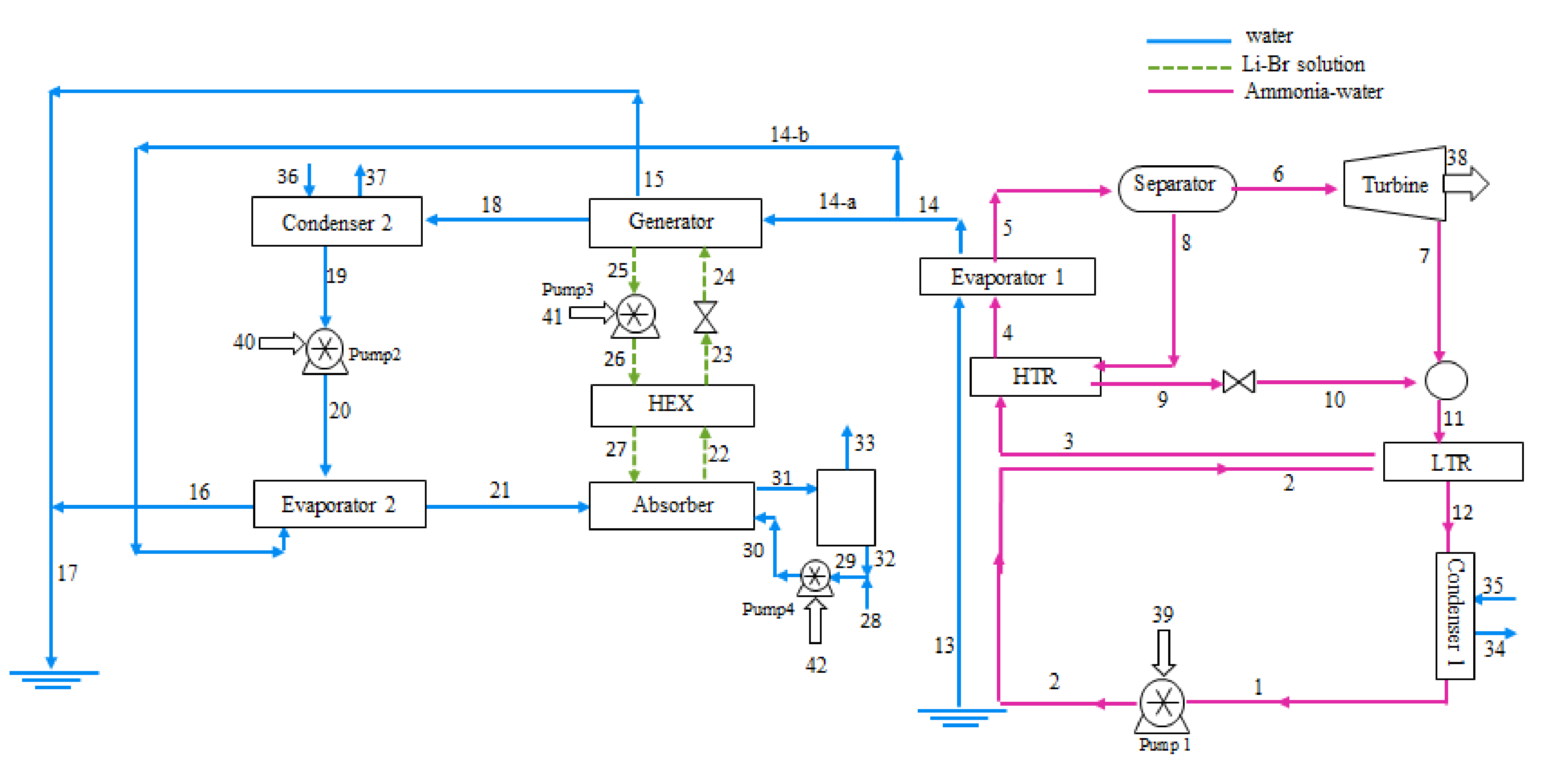
	Present work	[21]
P(bar)	4.769	4.6
T(°C)	8	8
X(-)	0.00	0.00
XI(-)	0.82	0.82
M(kg/s)	16.8	16.8

	Present work	[21]
P(bar)	5.6	5.6
T(°C)	29.9	30
X(-)	0.554	0.58
XI(-)	0.82	0.82
M(kg/s)	16.8	16.8

	Present work	[21]
P(bar)	5.6	5.6
T(°C)	29.9	30
X(-)	0.554	0.58
XI(-)	0.82	0.82
M(kg/s)	16.8	16.8

Combined Cycle

Combined → Kalina cycle + LiBr/H_2O absorption heat transformer



Thermodynamic analysis of combined cycle

subsystems	Exergy relations	Energy relations
Kalina cycle		
Evaporator 1	$E_{D,eva1} = T_0\left[m_4(s_5 - s_4) + m_{13}(s_{14} - s_{13})\right]$	$m_4(h_5 - h_4) = m_{13}(h_{14} - h_{13})$
Separator	$E_{D,sep} = T_0\left[m_6 s_6 + m_8 s_8 - m_5 s_5\right]$	$m_5 x_5 = m_6 x_6 + m_8 x_8$
Turbine	$E_{D,Tur} = T_0\left[m_6(s_6 - s_7)\right]$	$\eta_t = \frac{h_6 - h_7}{h_6 - h_{7s}}, w_t = m_6(h_6 - h_7)$
LTR	$E_{D,LTR} = T_0\left[m_{11}(s_{12} - s_{11}) + m_2(s_3 - s_2)\right]$	$m_2(h_3 - h_2) = m_{11}(h_{12} - h_{11})$
HTR	$E_{D,HTR} = T_0\left[m_3(s_4 - s_3) + m_8(s_9 - s_8)\right]$	$m_3(h_4 - h_3) = m_8(h_9 - h_8)$
Pump 1	$E_{D,P1} = T_0\left[m_1(s_2 - s_1)\right]$	$w_{p,1} = v_2(h_2 - h_1)$
Condenser 1	$E_{D,Con1} = T_0\left[m_1(s_1 - s_{12}) + m_{34}(s_{35} - s_{34})\right]$	$Q_{cond,1} = m_1(h_1 - h_{12})$
LiBr/H_2O cycle		
Evaporator 2	$E_{D,eva2} = T_0\left[m_{22}(s_{23} - s_{22}) + m_{15}(s_{15} - s_{13})\right]$	$m_{13}(h_{13} - h_{16}) = m_{22}(h_{22} - h_{23})$
Absorber	$E_{D,Abs} = T_0\left[(m_{17}s_{17} - m_{23}s_{23} - m_{26}s_{26}) + m_{29}(s_{30} - s_{29})\right]$	$m_{30}(h_{30} - h_{29}) = m_{17}h_{17} - m_{23}h_{23} - m_{26}h_{26}$
HEX	$E_{D,eva2} = T_0\left[m_{17}(s_{18} - s_{17}) + m_{25}(s_{26} - s_{25})\right]$	$m_{17}(h_{17} - h_{18}) = m_{25}(h_{25} - h_{26})$
Generator	$E_{D,Abs} = T_0\left[(m_{20}s_{20} - m_{24}s_{24} - m_{19}s_{19}) + m_{14}(s_{14} - s_{13})\right]$	$m_{13}(h_{13} - h_{16}) = m_{19}h_{19} - m_{20}h_{20} - m_{24}h_{24}$
Th. valve	$E_{D,V} = T_0\left[m_{24}(s_{25} - s_{24})\right]$	$m_{18}h_{18} = m_{19}h_{19}$
Pump 2	$E_{D,P2} = T_0\left[m_{21}(s_{22} - s_{21})\right]$	$w_{p,2} = v_{21}(h_{22} - h_{21})$
Pump 3	$E_{D,P3} = T_0\left[m_{24}(s_{25} - s_{24})\right]$	$w_{p,3} = v_{24}(h_{25} - h_{24})$
Pump 4	$E_{D,P4} = T_0\left[m_{28}(s_{29} - s_{28})\right]$	$w_{p,4} = v_{28}(h_{29} - h_{28})$
Condenser 2	$E_{D,Con2} = T_0\left[m_{20}(s_{21} - s_{20}) + m_{35}(s_{36} - s_{35})\right]$	$Q_{cond,2} = m_{20}(h_{20} - h_{21})$

Performance evaluation

First law efficiency

$$\eta_{\mathrm{I}} = \frac{\dot{W}_{net} + \dot{Q}_{abs}}{\dot{Q}_{in}}$$

$$\dot{W}_{net} = \dot{W}_{Tur} - (\dot{W}_{P,1} + \dot{W}_{P,2} + \dot{W}_{P,3} + \dot{W}_{P,4})$$

$$\dot{Q}_{abs} = m_{30}(h_{31} - h_{30})$$

$$\dot{Q}_{in} = m_{1}(h_{1} - h_{17})$$

Second law efficiency

$$\eta_{\mathrm{II}} = \frac{\dot{W}_{net} + \dot{E}_{abs}}{\dot{E}_{in}}$$

$$\dot{E}_{abs} = \dot{E}_{31} - \dot{E}_{30}$$

$$\dot{E}_{in} = m_{1}\left[(h_{1} - h_{17}) - T_{0}(s_{1} - s_{17})\right]$$

Thermoeconomic analysis

$$\sum \dot{C}_{out,k} + \dot{C}_{\mathrm{w},k} = \sum \dot{C}_{\mathrm{in},k} + \dot{C}_{\mathrm{q},k} + \dot{Z}_{k}$$

$$\dot{Z}_{k} = \dot{Z}_{k}^{CI} + \dot{Z}_{k}^{OM}$$

$$\dot{Z}_{k}^{CI} = (\frac{CRF}{\tau})Z_{k}$$

$$CRF = \frac{i_{r}(1+i_{r})^{n}}{(1+i_{r})^{n} - 1}$$

$$\dot{Z}_{k}^{OM} = \gamma_{k} Z_{k} + \omega_{k} \dot{E}_{P,k} + \dot{R}_{k}$$

The input data assumed in the simulation

Temperature of the Environment	25 $^{\circ}C$
Pressure of the Environment	1 bar
Temperature of the water from the well	124 $^{\circ}C$
Temperature of exit water of eveporator1	80 $^{\circ}C$
Turbine inlet pressure	32.3bar
Temperature of the water to the well	T_{14}-5
Temperature of the solution exit from the condenser	T_0+5
Temperature of the Generator and eveporator2	T_{16}-3
Mass flow rate of geothermal water	89 kg/s
Temperature of LiBr/H_2O solution	110 $^{\circ}C$
Mass flow rate of seawater	12 kg/s
Ammonia mass fraction	82%
Turbine isentropic efficiency	90%
Pump isentropic efficiency	80%

Thermodynamic properties and cost of streams for the combined cycle

state	t (C)	P (bar)	x	m (kg/s)	$\dot{E}_{ph}$ (kJ/kg)	$\dot{E}_{ch}$ (kJ/kgk)	$\dot{E}$	Costs $\dot{C}$	c
1	20	7.124	0	17.82	3,100	289,132	292,231	2455	2.333
2	20.6	32.3	-	17.82	3,164	289,132	292,295	2455	2.333
3	44.6	32.3	-	17.82	3,214	289,132	292,345	2457	2.335
4	65.6	32.3	-	17.82	3,382	289,132	292,513	2460	2.337
5	118	32.3	0.6824	17.82	6,388	289,132	295,520	2480	2.331
6	118	32.3	1	12.16	5,915	233,147	239,065	2007	2.332
7	46.4	7.124	0.9417	12.16	3,212	233,147	236,359	1984	2.332
8	118	32.3	0	5.658	470.4	55,984	56,455	475.4	2.339
9	49.6	32.3	-	5.658	170.8	55,984	56,155	472.9	2.339
10	50	7.124	-	5.658	154.5	55,984	56,139	472.7	2.339
11	49.6	7.124	0.6382	17.82	3,364	289,132	292,496	2457	2.333
12	40.4	7.124	0.5778	17.82	3,228	289,132	292,359	2456	2.333
13	124	2.25	-	89	5,085	0	5,085	23.8	1.3
14	80	2.25	-	89	1,689	0	1,689	7.906	1.3
14-a	80	2.25	-	40.89	913.2	0	913.2	4.274	1.3
14-b	80	2.25	-	48.11	776	0	776	3.632	1.3
15	75	2.25	-	40.89	647.4	0	647.4	3.03	1.3
16	75	2.25	-	48.11	761.8	0	761.8	3.565	
17	75	2.25	-	89	1,409	0	1,409	6.595	1.3
18	72	0.04246	-	0.4029	18.74	0	18.74	4.012	59.48
19	30	0.04246	-	0.4029	0.07032		0.07032	0.01506	59.48
20	30	0.3397	-	0.4029	0.08235	0	0.08235	0.02232	75.29
21	72	0.3397	-	0.4029	134.4	0	134.4	1.224	2.529
22	110	0.3397	0.5511	5.034	229.5	5.643	235.2	5.979	7.063
23	92.73	0.3397	0.5511	5.034	193.1	5.643	198.8	5.055	7.063
24	64.72	0.04246	0.5511	5.034	439.2	5.643	439.2	11.31	7.063
25	72	0.04246	0.5982	4.631	274.1	4.647	278.8	8.466	8.437
26	81.27	0.3397	0.5982	4.631	286.8	4.647	291.5	9.307	8.87
27	101.4	0.3397	0.5982	4.631	319.7	4.647	324.3	10.44	8.942
28	25	1	-	0.365	0.03545	0	0.03545	0	0
29	98.19	0.9494	-	15	488.1	0	488.1	20.4	11.61
30	98.19	1.013	-	15	488.3	0	488.3	20.41	11.61
31	100	1.013	-	15	676.6	0	676.6	27.19	11.15
32	100	1.013	-	14.67	498.6	0	498.6	20.4	11.36
33	100	1.013	-	0.365	178	0	178	8.255	12.82
34	15	1	0	677.5	485.2	0	485.2	0	0
35	20	1	-	677.5	119.6	0	119.6	3.28	7.617
36	15	1	-	48.33	34.61	0	34.61	0	0
37	20	1	-	48.33	8.532	0	8.532	4.246	138.2
38	-	-	-	-	-	-	2452	22.74	2.257
39	-	-	-	-	-	-	80.59	0.7473	2.256
40	-	-	-	-	-	-	0.01203	0.00011	2.576
41	-	-	-	-	-	-	83.04	0.7701	2.576
42	-	-	-	-	-	-	0.1108	0.00102	2.576

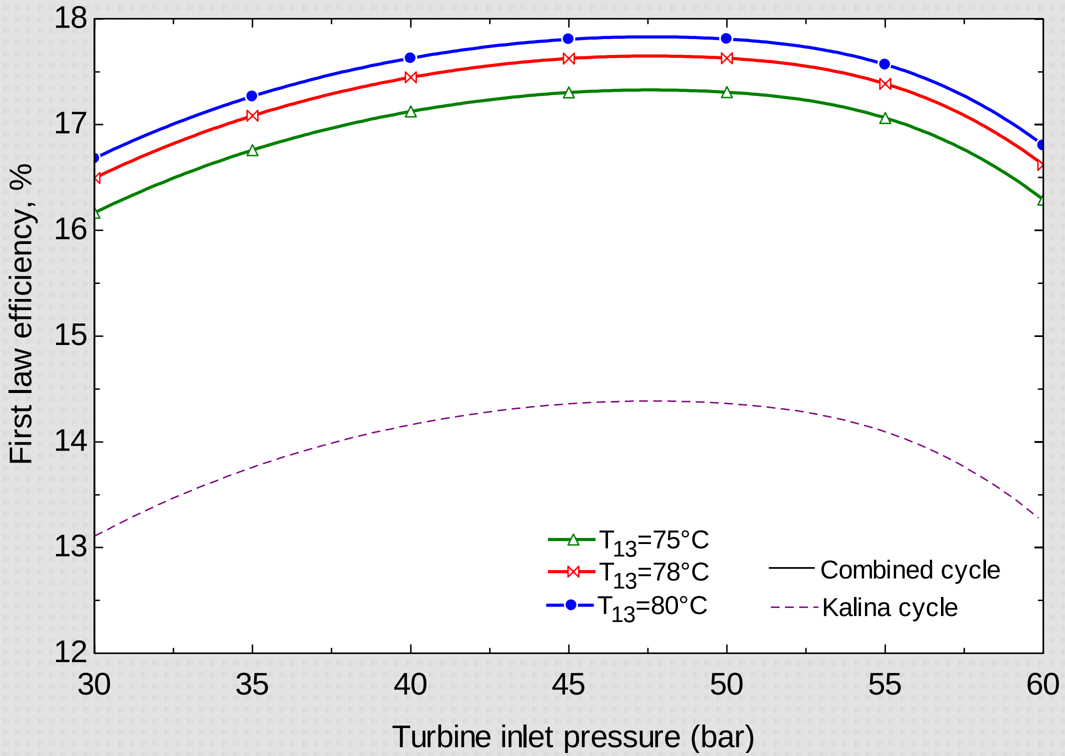
The performance of the combined cycle

Turbine work (kW)	2452
Condenser 1 heat rejection (kW)	14172
Pump 1 work (kW)	80.59
Pump 2 work (kW)	0.01203
Pump 3 work (kW)	83.04
Pump 4 work (kW)	0.1108
Evaporator 1 heat input (kW)	16543
Evaporator 2 heat input (kW)	1009
Absorber heat transfer (kW)	938.3
Generator heat transfer (kW)	857.3
Condenser 2 heat rejection (kW)	1011
Net power output of Kalina(kW)	2371
Net power output and absorber heat (kW)	3226
Heat input (kW)	18409
Exergy input (kW)	3676
Thermal efficiency (%)	17.52
Exergy efficiency (%)	67.38

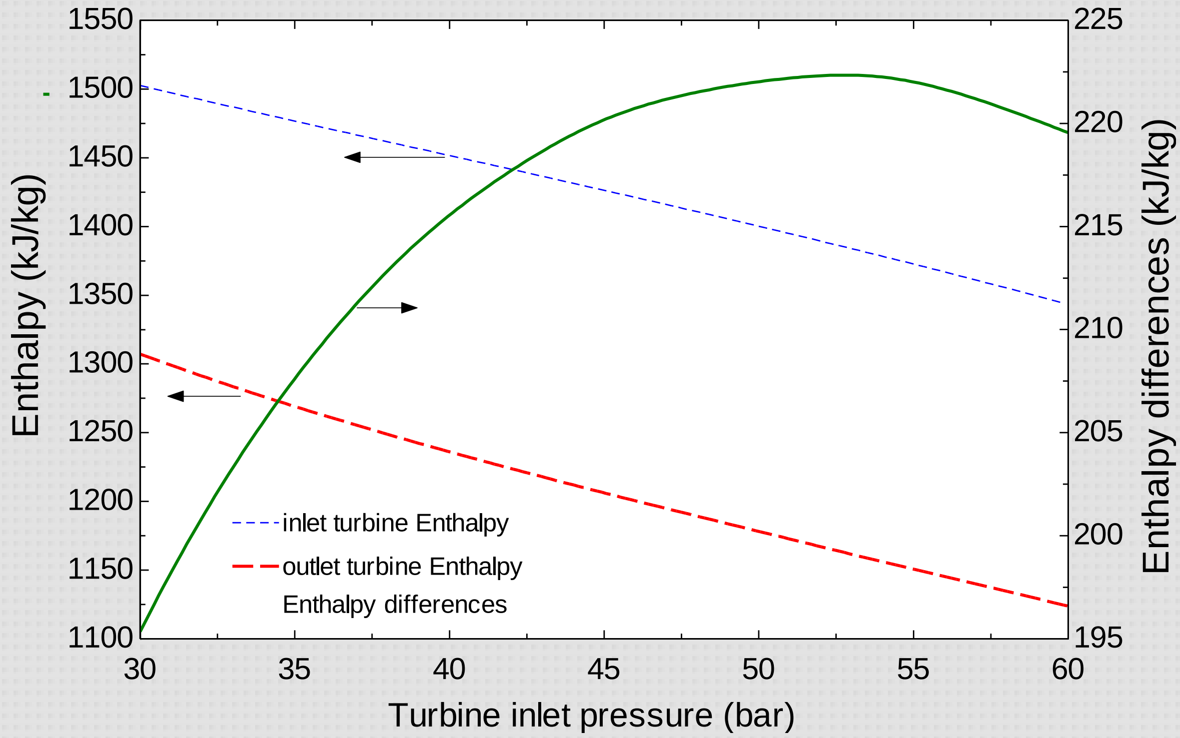
Cost analysis result for combined cycle

subsystems	$\dot{E}_{F,k}$ (kW)	$\dot{E}_{P,k}$ (kW)	$\dot{E}_{D,k}$ (kW)	$Y_{D,k}$ (%)	$Y^*_{D,k}$ (%)	ε_k (%)
Kalina cycle						
Evaporator 1	3396	3007	389	4.71	24.46	88.54
Turbine	2706	2452	254	3.06	15.97	90.61
LTR	137	50	87	1.04	5.47	36.49
HTR	300	168	132	1.59	0.1	56
Separator & valve	316	300	16	0.19	1.006	94.93
Pump 1	80.59	64	16.59	0.19	1.04	79.41
Condenser 1	364.6	128	236.6	2.85	14.88	35.1
LiBr/H_2O cycle						
Evaporator 2	134.31	14.2	118.31	1.42	7.44	10.57
Absorber	223.5	188.5	35	0.42	2.20	84.34
HEX	36.4	32.8	3.6	0.04	0.22	90.1
Generator	492.74	265.8	226.94	2.73	14.27	53.94
Pump 2	0.01204	0.01203	0.0001			
Pump 3	83.04	12.7	70.34	0.84	4.42	15.3
Pump 4	0.1108	0.11	0.0008			
Condenser 2	26.078	18.66	4.418	0.05	0.27	71.55
Overall system	8296.4	6701.8	1589.8	19.16	100	80.77

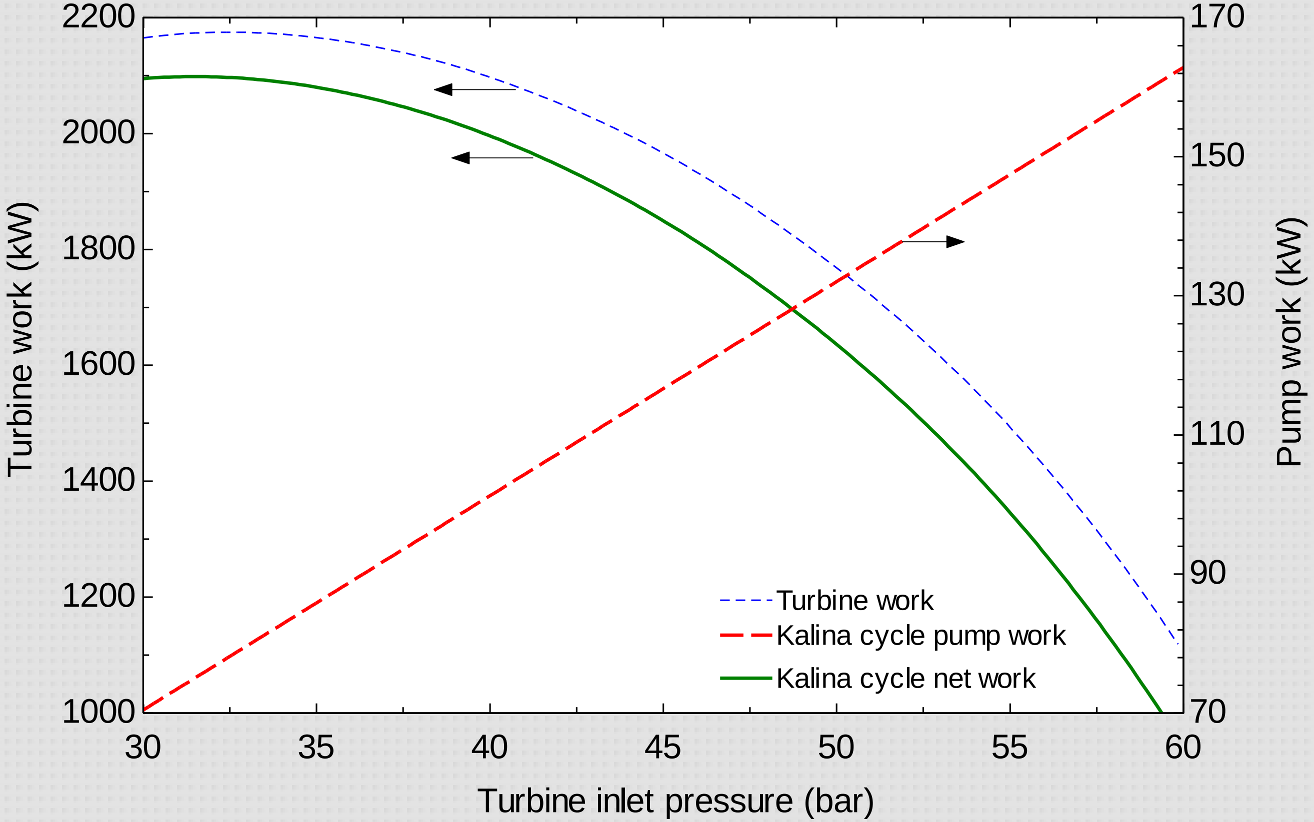
The effect of turbine inlet pressure on the Kalina and combined cycle energy efficiency for different evaporator exit temperature



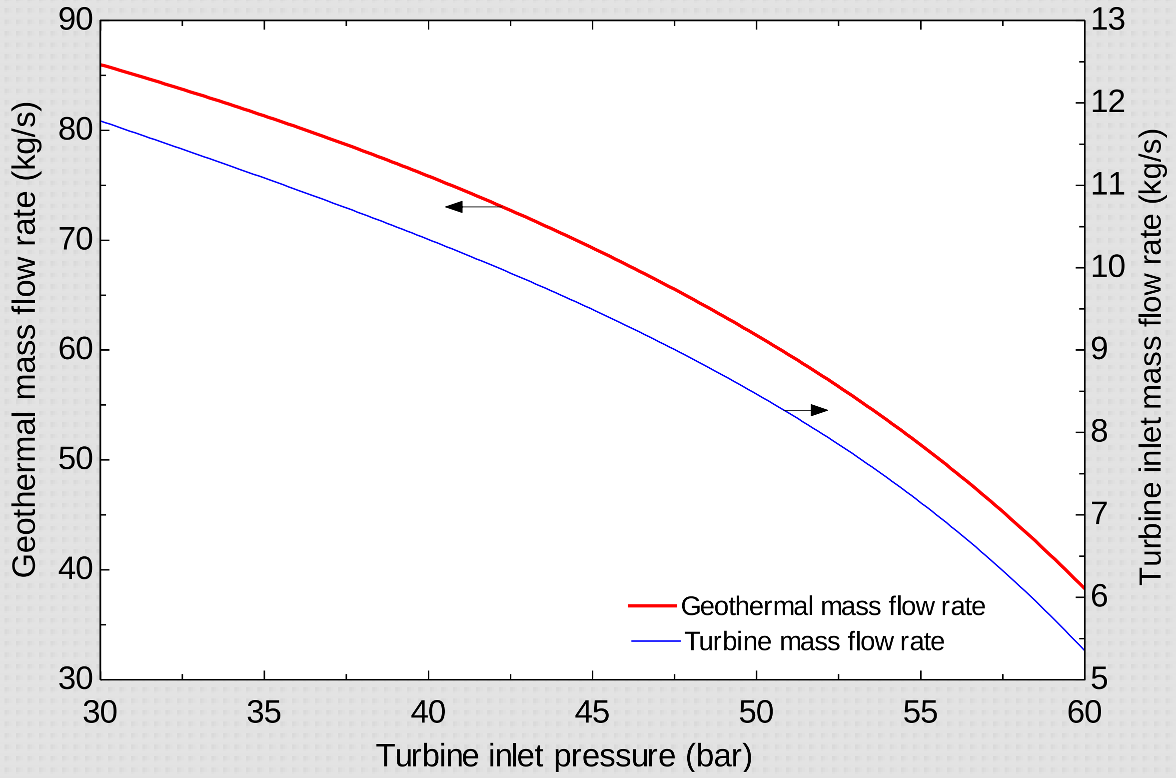
The effect of turbine inlet pressure on the turbine inlet and outlet enthalpy and their differences



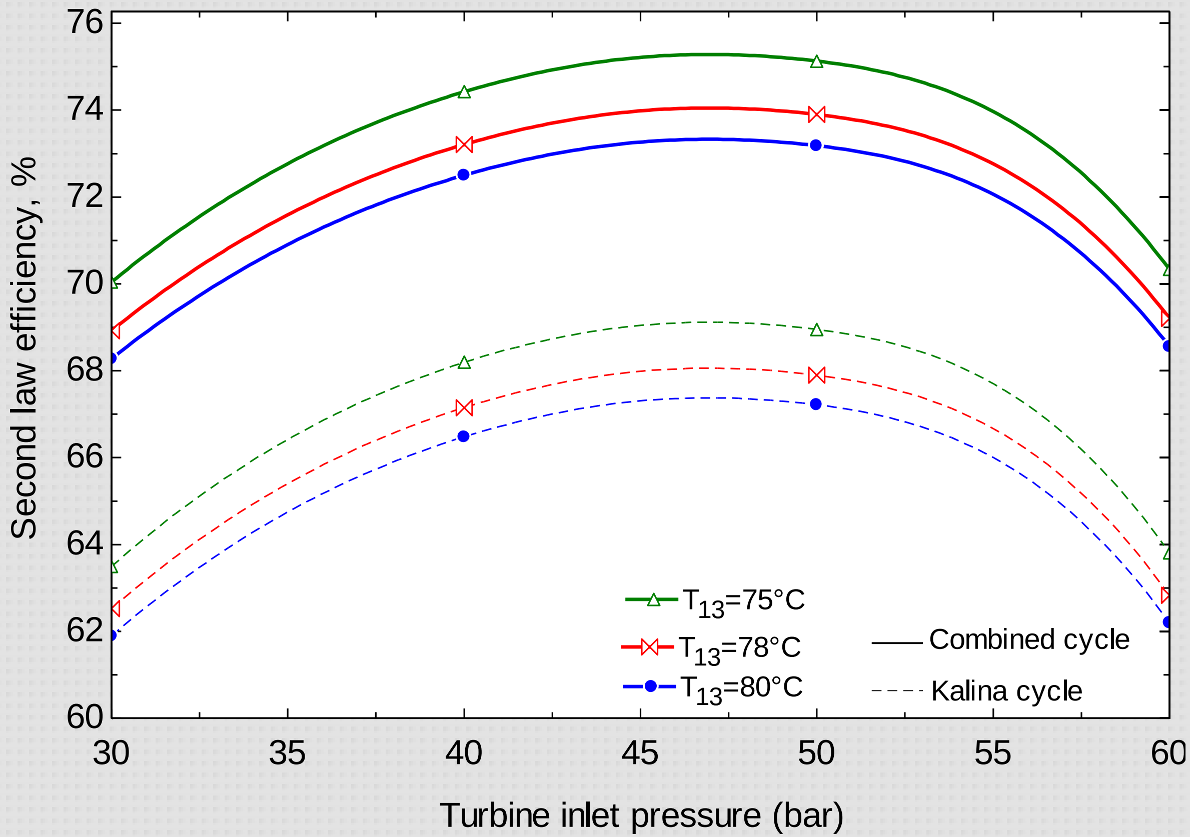
The effect of turbine inlet pressure on the cycle work



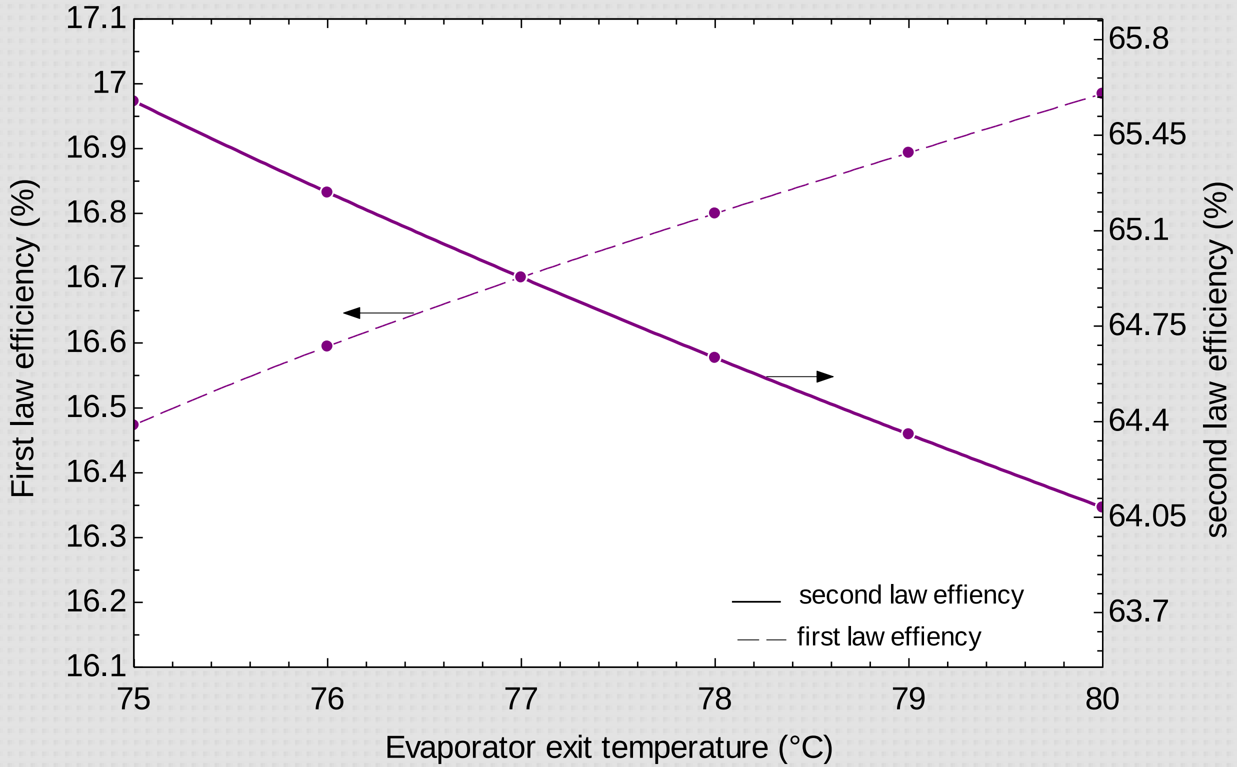
The effect of turbine inlet pressure on the geothermal and turbine inlet mass flow rate



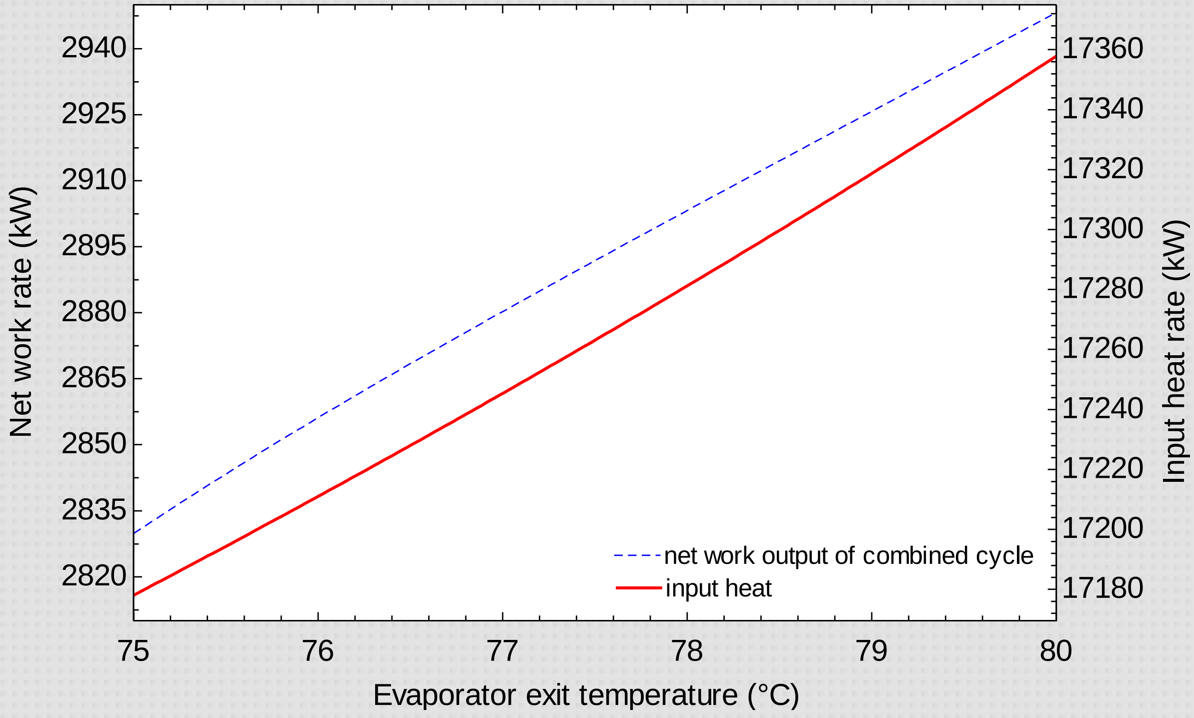
The effect of turbine inlet pressure on the Kalina and combined cycle exergy efficiency for different evaporator exit temperature.



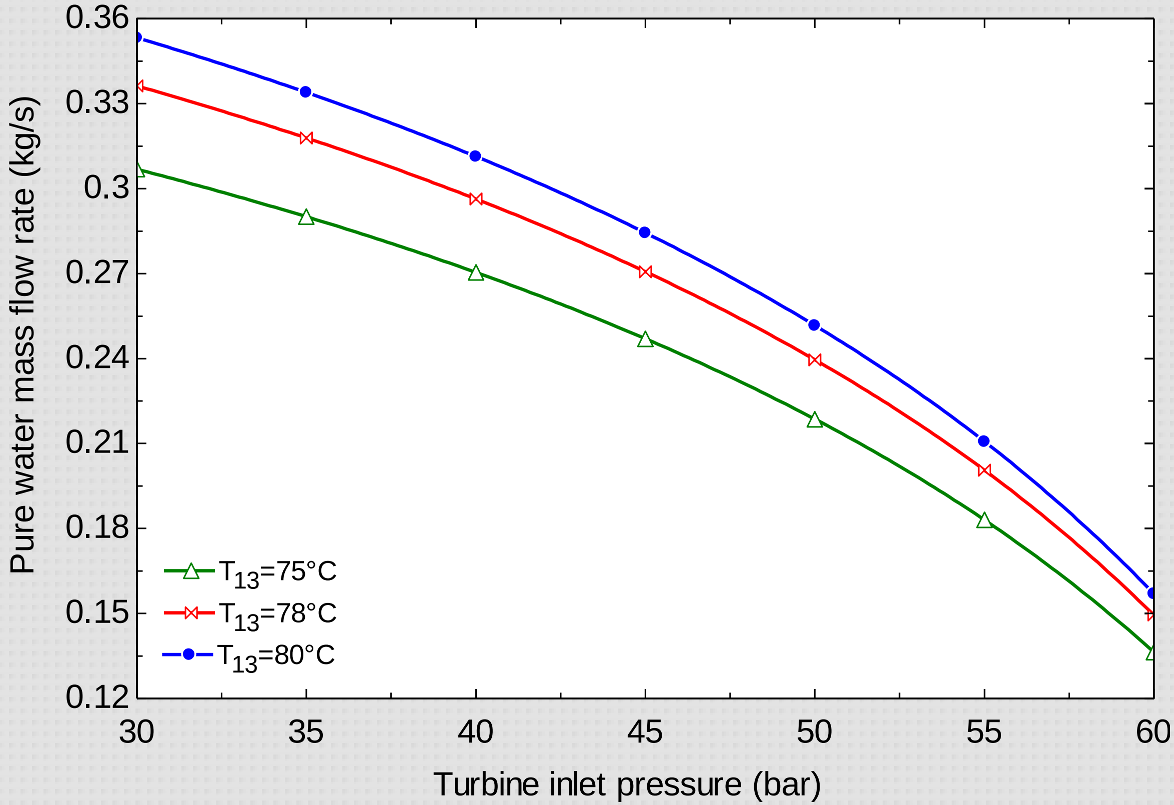
The effect of evaporator exit temperature on first and second law efficiency



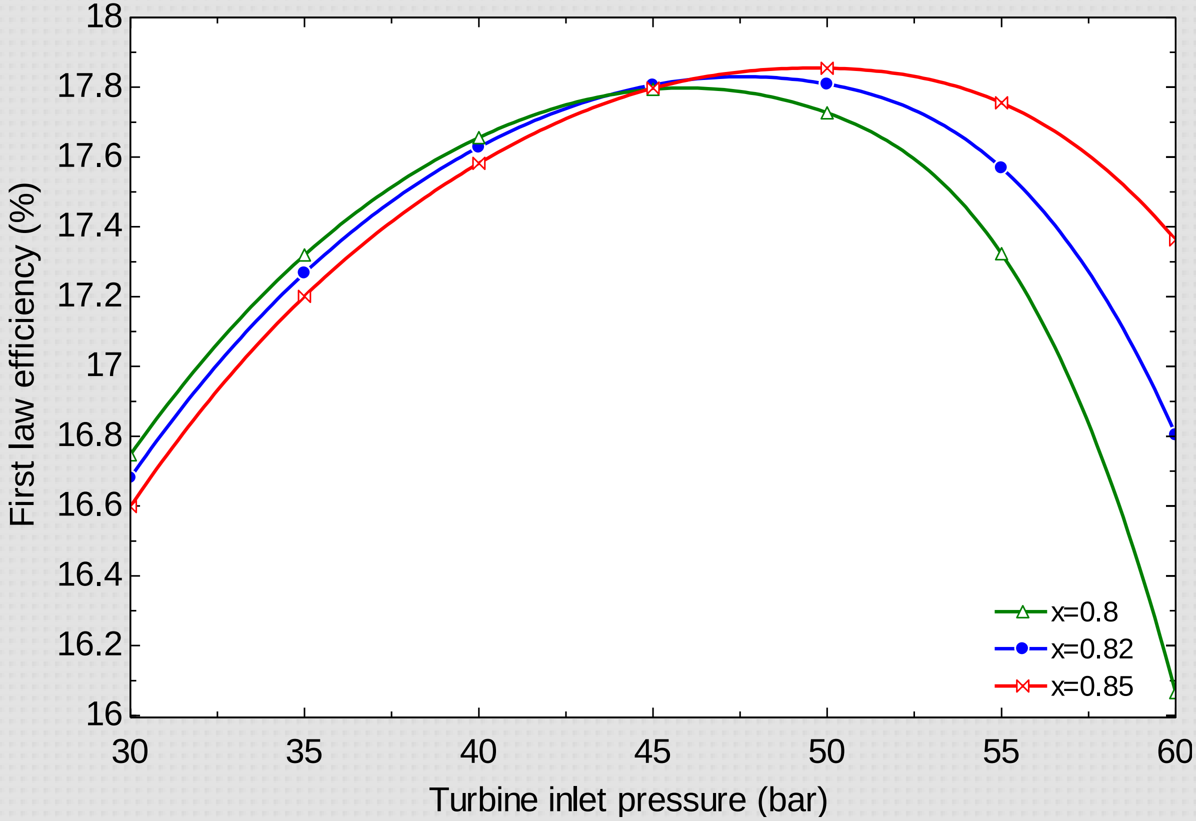
The effect of evaporator exit temperature on network and input heat of combined cycle



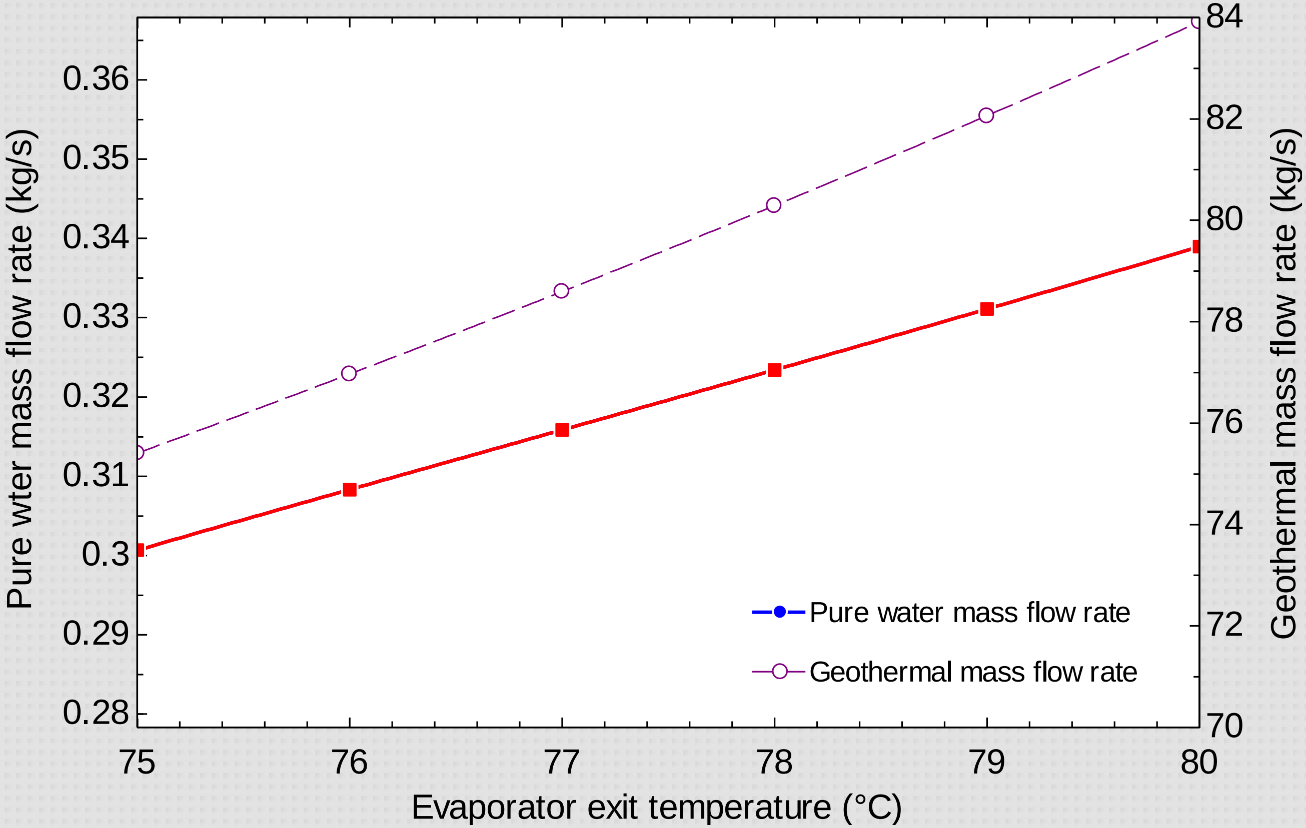
The effect of turbine inlet pressure on the produced pure water



The effect of turbine inlet pressure on the first law efficiency for several values of ammonia concentration



The effect of temperature of hot water exiting the evaporator1 on the pure water production



conclusion

- The proposed cycle which is a combination of Kalina cycle with ammonia – water as working fluid and a heat transformer cycle with lithium bromide – water as working fluid, can be a proper substitution for conventional geothermal power plants. The production of pure water by the proposed cycle is yet another advantage for the proposed cycle. The first and second law efficiencies of the proposed cycle are around 24% and 12.3% higher than the corresponding values for the Kalina cycle.
- The first and second law efficiencies are maximized at some particular values of pressure at the turbine inlet. The maximum values are increased with increasing ammonia concentration at evaporator1 outlet and increasing turbine inlet pressure.
- With increasing hot water temperature at the evaporator1 outlet,the first law efficiency increases and the second law efficiency decreases. However, higher temperature is suggested for the hot water exiting the evaporator1 as the second law efficiency is more meaningful criteria.
- As the turbine inlet pressure increases and/or hot water temperature at the exit of evaporatore1 decreases, the produced pure water mass flow rate decreases.
- The proposed cycle produces 2.94 MW and 0.34 kg/s pure water using geothermal water with a mass flow rate of 89 kg/s at a temperature of 124 ^{o}C.
- The evaporatore1 has the highest contribution in the cycle exergy destruction so that more attention is needed in design of this component.

Thanks for your attention