Performance Simulation of Combined Cycle with Kalina Bottoming Cycle

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ABSTRACT

The Kalina cycle has potential for improved performance regarding electrical efficiency, specific power output and cost of electricity compared with conventional technology because the mixture of working fluids enables efficient energy recovery. Thermodynamic analysis has been carried out for combined cycle with the Kalina bottoming cycle. In this work, the identified key parameters for the Kalina cycle are turbine inlet condition (pressure, temperature and concentration), separator temperature and ambient temperature. The effect of these parameters on exergy efficiency of combined cycle is examined. The combined cycle efficiency increases with the increase in the turbine inlet pressure, and the same decreases with increases in ambient temperature, turbine inlet temperature and its concentration. Heat recovery from exhaust decreases with increases in the separator temperature, and it does not alter the output of the combined cycle. The efficiency of the cycle is very sensitive to the turbine inlet concentration and ambient temperature.

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Keywords: ammonia-water vapor mixture, combined cycle, Kalina cycle, properties, thermodynamic analysis.

Nomenclature

AP	approach point, K
F	fraction of mass to be converted into vapor
h	specific enthalpy, kJ/kg mol
m	mass, kg/kg mol of fuel
M	molecular weight
P	pressure, bar
PP	pinch point, K
T	temperature, K
w	work, kJ/kg mol
x	molar fraction of ammonia in the liquid phase, kg mol/kg mol mixture
y	molar fraction of ammonia in the vapor phase, kg mol/kg mol mixture
c	molar fraction of ammonia in the liquid-vapor equilibrium, kg mol/kg mol
ξ	mass fraction of ammonia in the liquid phase, kg/kg mixture
ζ	mass fraction of ammonia in the vapor phase, kg/kg mixture
η	efficiency
ε	specific exergy at ground state, kJ/kg mol

Subscripts

c	compressor
cc	combined cycle
gc	gas cycle
gt	gas turbine
kc	Kalina cycle
k	Kalina
p	pump

INTRODUCTION

Ammonia water mixtures have been used in the industry for over 100 years as a working fluid for refrigeration systems. Its use in power cycles is relatively recent. The Kalina cycle is a thermodynamic cycle for converting thermal energy to mechanical power, which utilizes a working fluid comprised of at least two different components. A ratio between those components is varied in different parts of the system to increase thermodynamic reversibility and therefore increase overall thermodynamic efficiency. Improving energy efficiency lowers the implicit price of energy and hence makes its use more affordable [1]. The Kalina cycle was first invented by the Russian Engineer Aleksandr Kalina [2]. A Rankine cycle, with steam as the working fluid, is usually the bottoming cycle of a gas-based combined cycle power plant. In a steam-based cycle, boiling essentially takes place at constant pressure and temperature and does not have a good match with the temperature profile of the exiting flue gas from the topping cycle, resulting in the large loss of availability. When a binary water-ammonia mixture changes its state from liquid to vapor, the more volatile ammonia vaporizes first and then the water starts vaporizing. Hence, a better match with the temperature profile of the flue gas ensues. Basically, this concept is suitable for medium to low gas temperature heat recovery systems with gas inlet temperatures in the range of between 200 and 525°C, offering more gains (over Rankine cycle) as the gas temperature decreases.

Sayed and Tribus [3] made a theoretical comparison of the Kalina cycle with the Rankine cycle. The configurations developed by them were very complicated because several heat exchangers had more than two streams. Later, Marston [4] modified the Sayed and Tribus configuration with simple two stream heat exchangers, and performed the thermodynamic analysis by using the property charts. Rogdakis and Antonopolos [5] proposed an absorption power cycle with ammonia-water mixture as a working agent and compared the efficiency with the Rankine cycle. This cycle was also optimized by Rogdakis [6], who developed correlations describing the optimum operation of the cycle. Nag and Gupta [7] presented an exergetic analysis of a modified Kalina cycle. Borgert and Velasquez [8] proved the economic advantage for a cogeneration plant with the use of an ammonia-water Kalina absorption power cycle as a bottoming cycle. Thermodynamic properties of ammonia-water system developed by Patek and Klomfar [9] are used in this work. They developed a set of five equations describing vapor-liquid equilibrium properties of the ammonia-water system. These properties cover the region within which absorption cycles commonly used operate most often. In 1992, a Kalina demonstration plant started operation at the U.S. Department of Energy's Energy Technology Engineering Center in California. At first the plant used waste heat at a temperature of approximately 540°C to generate 3 MWe of power and accumulated 5,200 operation hours. In this plant, the maximum pressure and temperature of the Kalina cycle were 110 bar and 516°C [10]. The Kalina cycle turbine was less expensive than the steam turbine, because the volumetric flow rate in the low pressure part of the ammonia-water turbine was much smaller than in the steam turbine, while the Kalina cycle heat exchangers were more expensive. However, the additional power output of the Kalina cycle gave an economic benefit compared with the steam cycle.

The main objective of this work is to examine the effect of the Kalina cycle operating parameters on the exergy efficiency of combined cycle to maximize the performance. Combined cycle with Kalina bottoming is compared with the conventional steam based combined cycle with respect to the topping cycle parameters.

THERMODYNAMIC ANALYSIS OF THE KALINA CYCLE

Assumptions used in the thermodynamic analysis:

1. Atmospheric condition is 1.01325 bar and 25°C.
2. Pressure ratio of gas cycle is 15
3. Gas cycle maximum temperature is 1200°C.
4. Ammonia mixture turbine inlet condition is 100 bar and 500°C.
5. Pinch point (PP) in the heat recovery steam generator (HRSG) is 5.0
6. Approach point (AP) in condenser, absorber, reheater and feed water heater is 5.0.
7. In the separator, the vapor portion is completely separate.
8. The isentropic efficiency for the air compressor and turbine in the Kalina cycle is 85%.
9. The isentropic efficiency of the gas turbine is 90%.
10. The condensate leaving the condenser is saturated liquid.
11. Pressure drop and heat loss in pipe lines are negligible.

The schematic flow diagram shown in Figure 1 illustrates a combined cycle with a simple Brayton topping cycle coupled with the Kalina cycle. The turbine exhaust (14) of the Kalina cycle is cooled in the distiller, then diluted with a weak solution (4) and condensed in the absorber. The saturated liquid leaving the absorber is heated in a distiller by the turbine exhaust (21). The working fluid is separated into

into rich ammonia water vapor (5) and poor liquid mixture (2) in the separator. The rich vapor mixture is mixed with the condensate from the splitter to obtain the desired concentration (7). It is then condensed in a condenser, compressed, heated in a feed water heater and sent to the boiler (HRSG), where it is superheated by the exhaust from the gas turbine (25). The superheated vapor (13) is expanded in the turbine, thus providing output from the Kalina cycle. Rich ammonia vapor from the separator cools in the feed water heater.

The Kalina cycle is solved with 1 kg mixture at the turbine inlet. The problem is extended with the combined cycle having 1 kg mol of fuel in the combustion chamber. The properties and mass flow rates around the loop are calculated by mass, concentration and energy balance equations. In the separator, both the liquid and vapor quantities are separated.

Figure 2 represents the temperature-enthalpy diagram of the Kalina bottoming cycle for combined cycle operating under the conditions y_{13} = 0.6, T_{13} = 500°C and T_1 = 70°C. The characteristic states of the working fluid are represented by the same numbers as those of the schematic diagram in Figure 1.

The molar fraction of ammonia in vapor phase from the given mass fraction in the vapor phase at the inlet of turbine, (x and ξ are used in the case of liquid phase).

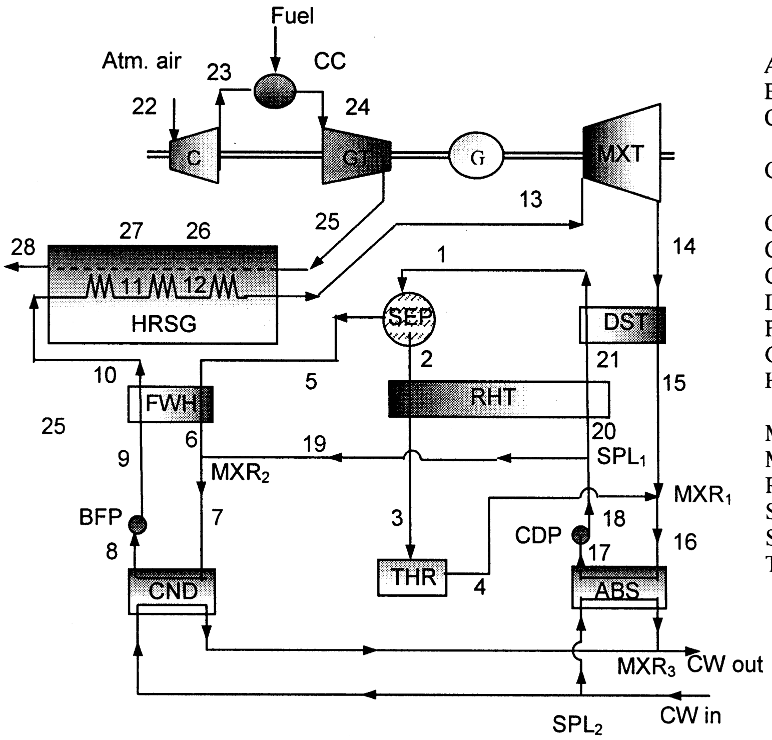
$$y_{13} = \frac{M_{NH_3}}{M_{H_2O} - \zeta_{13}} = \frac{17\zeta_{13}}{18 - \zeta_{13}} \qquad (1)$$

The fraction of mass to be converted into vapour (F) at the separator for the given separator temperature is determined with the simultaneous solution of mass and concentration balance equations. These equations are applied for separator, MXR_1 and MXR_2.

$$F = 1 - \sqrt{1 - y_{13}} \qquad (2)$$

The concentration after mixing in MXR_1, $c_{16} = \frac{y_{13}}{2 - F}$ (3)

The outlet temperature of the liquid mixture at the condenser and absorber is determined from the approach point and water inlet temperature. At this temperature and known concentration, intermediate



ABS	Absorber
BFP	Boiler Feed Pump
CC	Combustion Chamber
CDP	Condensate Feed Pump
CND	Condenser
CW in	Cooling Water in
CW out	Cooling Water out
DST	Distiller
FWH	Feed Water Heater
GT	Gas turbine
HRSG	Heat Recovery Steam Generator
MXR	Mixer
MXT	Mixture Turbine
RHT	Reheater
SEP	Separator
SPL	Splitter
THR	Throttle valve

Figure 1. Schematic flow diagram of the Kalina cycle

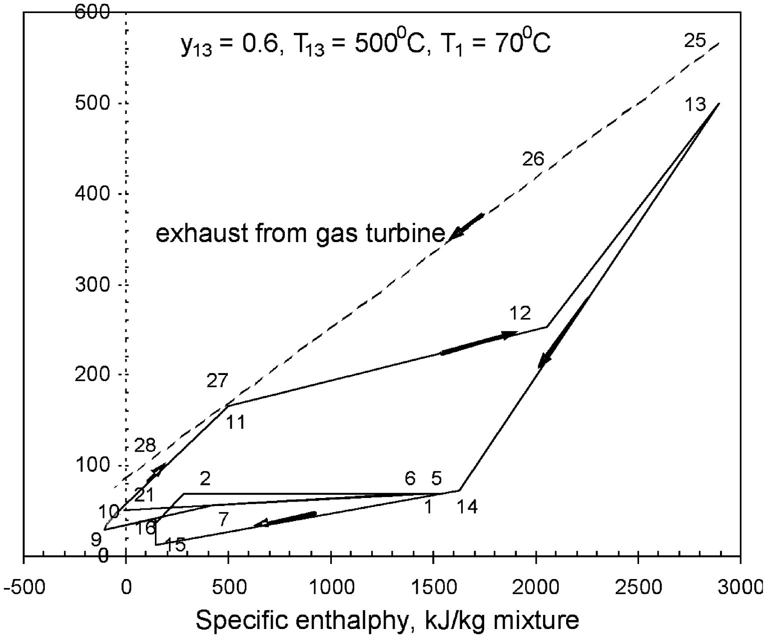


Figure 2. Temperature-enthalpy diagram of the Kalina bottoming cycle

pressure (IP) and low pressure (LP) are determined with the iteration. The working fluid is super heated in the HRSG with the heat recovery from exhaust. The mixture before entering into the separator is heated in the reheater. The outlet temperature of the separated liquid in the reheater is determined with the approach point. Similarly the outlet temperature of high pressure mixture in the feed water heater is determined with the approach point.

In the absorber the mixture is cooled to saturated liquid phase, therefore, $x_{17} = c_{16}$.

$$\text{In the reheater, } T_3 = T_{20} + AP \tag{4}$$

$$\text{In the feed water heater, } T_{10} = T_5 + AP \tag{5}$$

The outlet temperature of the flue gas from the HRSG,

$$T_{27} = T_{11} + PP \tag{6}$$

From heat balance in HRSG, the working fluid in the Kalina cycle,

$$m_k = \frac{h_{25} - h_{27}}{h_{13} - h_{11}} \text{ kg/kg mol fuel} \quad (7)$$

Gas turbine output, $w_{gt} = (h_{24} - h_{25})$ kJ/kg mol fuel (8)

Air compressor input, $w_c = (h_{23} - h_{22})$ kJ/kg mol (9)

The net output from gas cycle, $w_{net,\,gc} = (w_{gt} - w_c)$ kJ/kg mol (10)

In Kalina cycle, turbine output, $w_k = m_k\,(h_{13} - h_{14})$ kJ/kg mol fuel (11)

Work input for both BFP and CDP,

$$w_p = m_k\,(h_{18} - h_{17} + h_9 - h_8) \text{ kJ/kg mol} \quad (12)$$

Net work output from Kalina cycle,

$$w_{net,\,kc} = (w_k - w_p) \text{ kJ/kg mol} \quad (13)$$

Total output from the combined cycle,

$$w_{net,\,cc} = (w_{net,\,gc} + w_{net,\,kc}) \text{ kJ/kg mol} \quad (14)$$

To determine the exergy efficiency of the combined cycle, the exergy value of methane as fuel (ε^0) is taken as 836,420 kJ/kg mol [11].

Exergy efficiency of the combined cycle, $\eta_2 = \frac{w_{net\,cc}}{\varepsilon^0_{CH4}} \times 100$ (15)

RESULTS AND DISCUSSION

The advantage of the Kalina bottoming cycle over the steam bottoming cycle is compared for the combined cycle. The effect of Kalina cycle parameters, i.e., turbine inlet condition (pressure, temperature and concentration), separator temperature and ambient temperature on the exergy efficiency of the combined cycle has been studied. The net output of the combined cycle in percent of exergy of fuel is expressed as exergy efficiency of the combined cycle.

Figure 3 compares the two systems of the combined cycle; one is the combined cycle with the steam bottoming cycle having triple pressure HRSG and the other is the Kalina bottoming cycle. The comparison is made with variation in the pressure ratio of the air compressor at a gas turbine inlet temperature of 1200°C. The exergy efficiency of the Kalina bottoming combined cycle is higher than the efficiency with steam bottoming combined cycle. At 15 pressure ratio, the efficiency of the Kalina based combined cycle is 3.0 percent higher than the steam based combined cycle at the turbine inlet concentration of 0.6.

Figure 4 compares the two models of the combined cycle, with respect to the gas turbine inlet temperature at the pressure ratio of 15. The Kalina based cycle has 2.0 percent excess efficiency compared to combined cycle with steam bottoming at 1300°C. Based on these two results, adding the Kalina cycle to the gas cycle improves the efficiency compared with the case combined cycle with steam bottoming cycle.

Figure 5 depicts the effect of separator temperature on efficiency of the cycle. The exhaust temperature is also plotted in the same figure because the separator temperature governs the heat recovery from the exhaust. The LP and IP pressures do not alter with the separator tem-

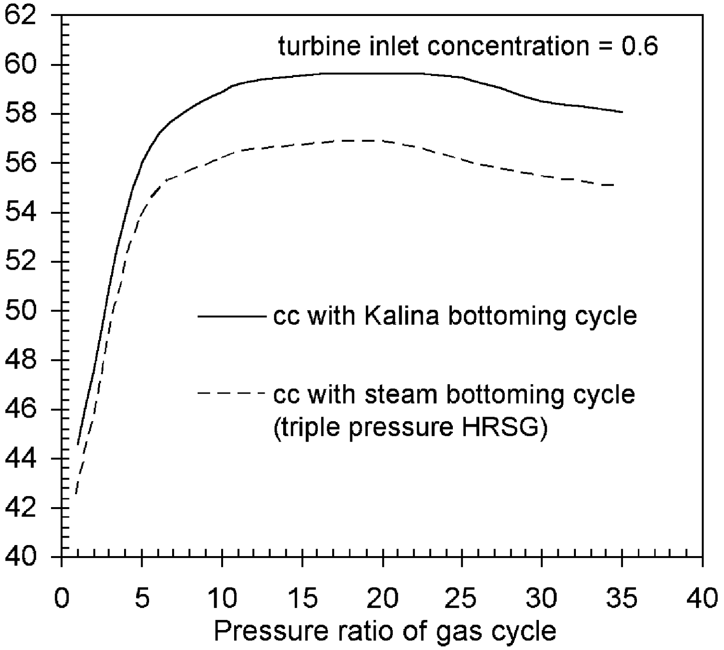


Figure 3. Comparison of combined cycle with Kalina bottoming and Rankine bottoming (with triple pressure HRSG) cycle with respect to the pressure ratio of the gas cycle

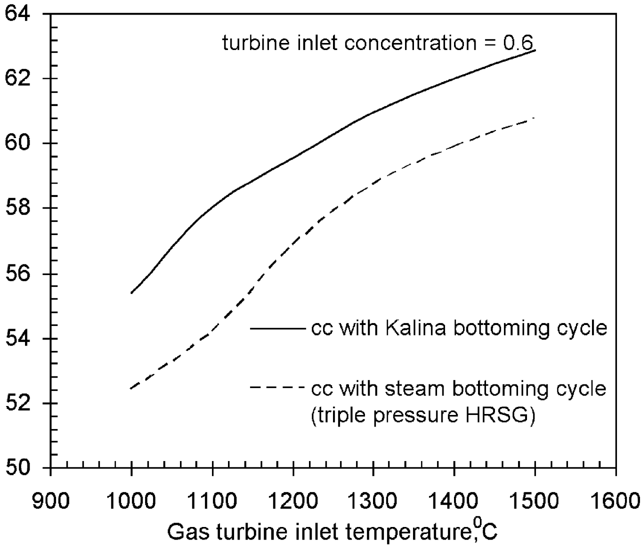


Figure 4. Comparison of combined cycle with Kalina bottoming and Rankine bottoming (with triple pressure HRSG) cycle with respect to the gas turbine inlet temperature

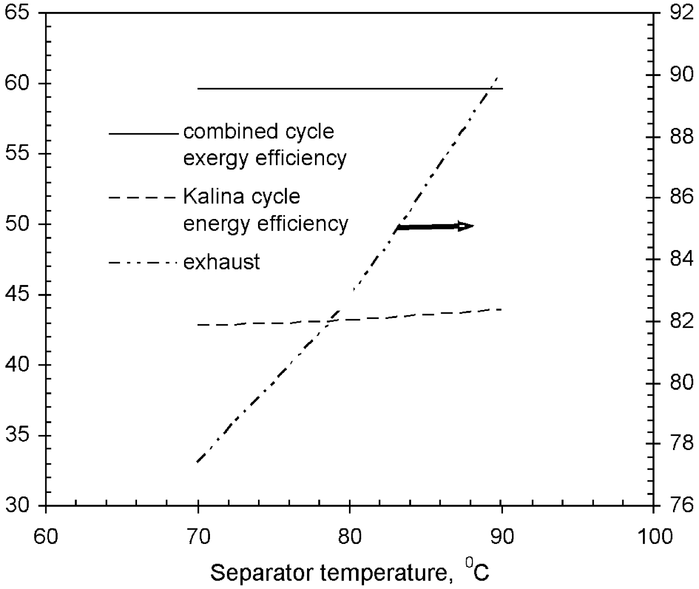


Figure 5. Effect of separator temperature on efficiency of cycle and exhaust temperature

perature. So the output from the Kalina turbine is fixed with the separator temperature. The heat input to the Kalina cycle decreases because of an increase in the exhaust gas temperature, which rises with increases in separator temperature. The energy efficiency of Kalina cycle, determined with the heat input to the Kalina cycle increases with an increase in the separator temperature. But because of constant output from turbines, combined cycle efficiency remains same.

Figure 6 shows the effect turbine inlet pressure and concentration have on the combined cycle efficiency. The efficiency increases with increases in the pressure and decreases with increases in the concentration. The LP pressure increases with increases in the ammonia concentration at the turbine inlet. It restricts the expansion of the mixture in the turbine so the output from the turbine decreases. The net effect decreases the efficiency with increases in the ammonia concentration at turbine inlet. The expansion in the turbine increases with increases in the inlet pressure, which results in increased turbine output and increased cycle efficiency. The benefit of high pressure is obtained up to the concentra-

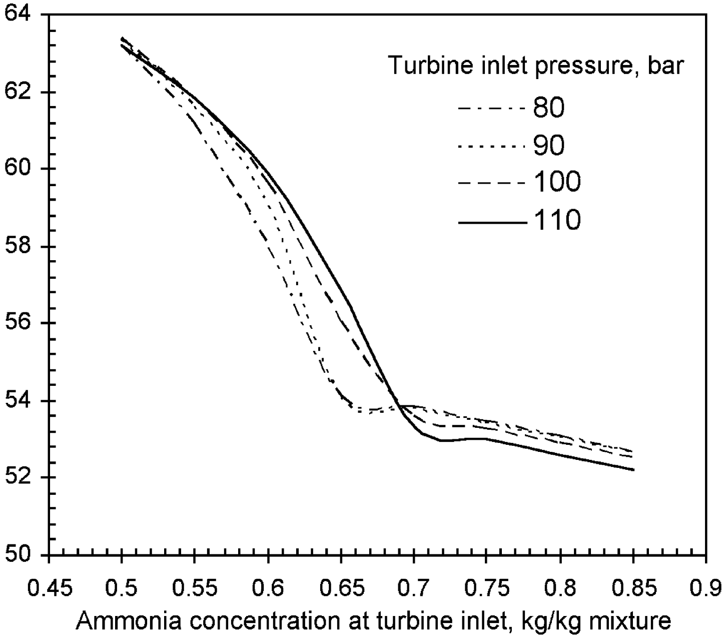


Figure 6. Effect of turbine inlet condition (pressure and concentration) on exergy efficiency of the combined cycle

tion of 0.7 only. For more than 0.7 concentrations, the working fluid after expansion becomes superheated which drops the output from the Kalina cycle.

Figure 7 shows the influence of turbine inlet temperature and concentration on exergy efficiency of the combined cycle. The efficiency of the cycle decreases with increases in both temperature and concentration. The amount of working fluid for the Kalina cycle generating in HRSG decreases with increases in the turbine inlet temperature. It results in decreased output from the Kalina cycle and also overall efficiency. The advantage of high temperature is obtained from the concentration of 0.7. But this efficiency is relatively low compared to the efficiency at lower concentration. Therefore, it is recommended to maintain the concentration level below 0.7.

Figure 8 shows the effect of ambient temperature on the LP and IP pressures in the Kalina cycle. The absorber (LP pressure) and condenser pressure (IP pressure) increases with increases in the ambient tempera-

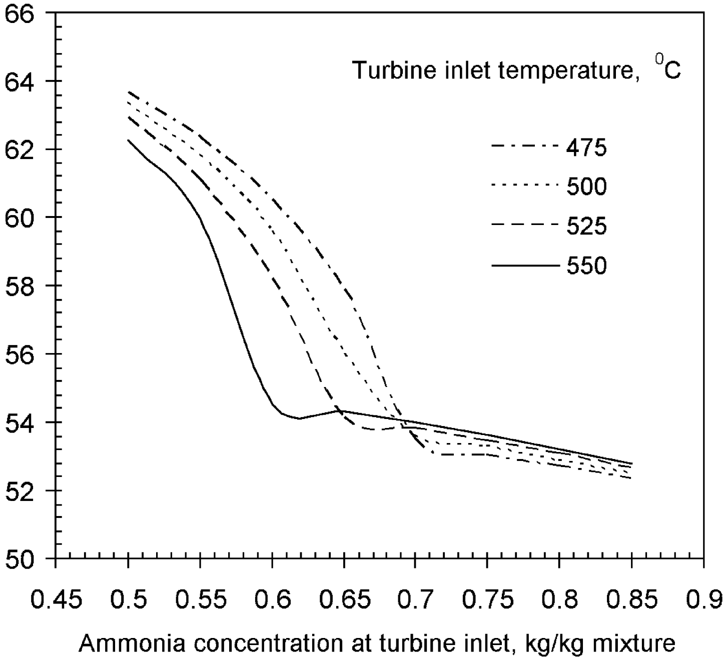


Figure 7. Effect of turbine inlet condition (temperature and concentration) on exergy efficiency of the combined cycle

ture. The expansion in the turbine decreases with increases in the condenser pressure which reduces the output from the Kalina cycle.

Figure 9 presents the effect of ambient temperature on the efficiency of the combined cycle. As mentioned before, the output of the Kalina cycle decreases with increases in the ambient temperature. It reduces the combined cycle efficiency. Therefore, the combined cycle efficiency is very sensitive with respect to the ambient condition.

CONCLUSION

Rankine bottoming cycle is replaced by the Kalina bottoming cycle in the combined cycle, and the merit of this alteration is proven with the comparison of the both cycles. The bottoming Kalina cycle parameters are identified as turbine inlet condition, separator temperature and ambient temperature. The Kalina cycle efficiency and hence the combined

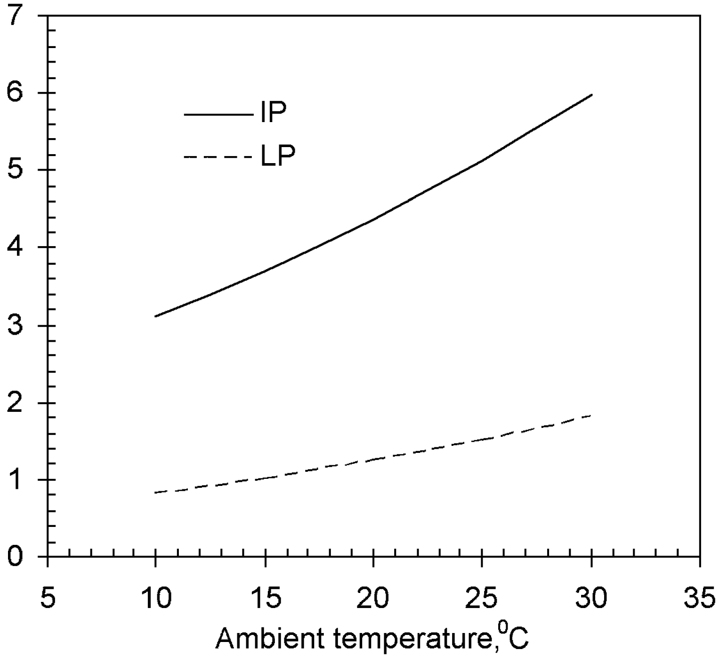


Figure 8. Influence of atmospheric temperature on the Kalina cycle LP and IP pressures

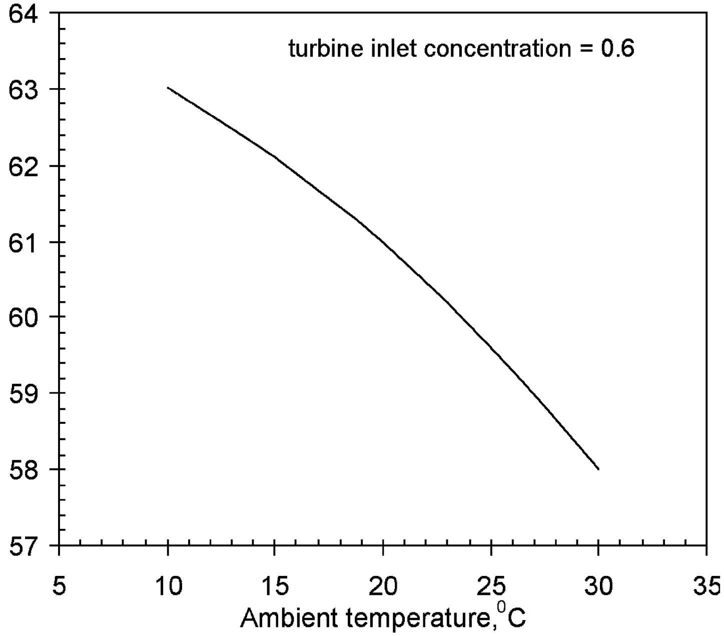


Figure 9. Influence of atmospheric temperature on exergy efficiency of combined cycle

cycle efficiency is very sensitive with the turbine inlet concentration and ambient condition. The exergy efficiency of the combined cycle increases with increases in the Kalina cycle turbine inlet pressure and decreases with increases in the ambient temperature, turbine inlet temperature and its concentration. The combined cycle efficiency remains the same, with the variation in the separator temperature. It is suggested to maintain the Kalina cycle turbine inlet concentration below 0.7 to get high combined cycle efficiency.

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APPENDIX

For calculating thermodynamic properties, both in the vapor and liquid phases, as well as in the vapor-liquid equilibrium, a simplified formulation proposed by Patek and Klomfar [9] is described by the following equations.

$$T(P,x) = T_0 \sum_i a_i (1-x)^{m_i} \left[ln\left(\frac{P_0}{P}\right)\right]^{n_i} \tag{A1}$$

$$T(P,y) = T_0 \sum_i a_i (1-y)^{m_i/4} \left[ln\left(\frac{P_0}{P}\right)\right]^{n_i} \tag{A2}$$

$$y(P,x) = 1 - exp\left[ln(1-x) \sum_i a_i \left(\frac{P}{P_0}\right)^{m_i} x^{n_i/3}\right] \tag{A3}$$

$$h_L(T,x) = h_0 \sum_i a_i \left(\frac{T}{T_0} - 1 \right)^{m_i} x^{n_i} \qquad \text{(A4)}$$

$$h_G(T,y) = h_0 \sum_i a_i \left(1 - \frac{T}{T_0} \right)^{m_i} (1 - y)^{n_i/4} \qquad \text{(A5)}$$

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