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Thermodynamic Analysis of Rankine-Kalina Combined Cycle

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Abstract

Efficiency enhancement in a low grade fuel fired power plant is one of the challenging tasks for researchers. In a low grade fuel fired power plant even a fraction of a percentage improvement in efficiency implies a huge savings in annual fuel costs. Mainly, the poor vapor quality of steam in the last stages of an LP turbine and energy loss in the condenser deteriorates the Rankine steam cycle performance. Reducing the amount of energy loss in the condenser and minimizing two-phase fluid operation in last stages of the LP turbine can substantially improve the cycle efficiency. The objective is to reduce the energy losses and to enhance the system performance. In this work a direct-fired 82.2 MW_{fuel} biomass fueled condensing power Rankine cycle is considered for performance improvement. Energy and exergy analysis are performed for the proposed Rankine-Kalina combined cycle (RKC). The RKC cycle produces higher power output and is more efficient than a Rankine steam cycle.

Keywords: Rankine-Kalina combined cycle, low grade fuel, biomass.

1. Introduction

The efficiency of the Rankine cycle can be improved by varying cycle parameters such as turbine inlet pressure, inlet temperature, reheat pressure, reheat temperature, extraction pressure and the condenser pressure with respect to the optimum value. The last few stages of an LP turbine usually operate in the two-phase region and they are subjected to blade corrosion problems. Mainly, blade erosion occurs due to sudden impingement of moisture droplets at the leading edge of the blades. The energy loss due to moisture reduces the power output and thus, plant profitability (Dooley, 2001). Specific volume of the steam is gradually increasing as the steam expands in the steam turbine. The substantial increase in specific volume in the LP turbine leads to careful design of LP turbine stages and exhaust part. Appropriate selection of blade material, and exhaust hood area are of paramount importance in design. (Li et al., 1985). The energy loss due to moisture and energy loss in the condenser are unavoidable losses in steam electric power plants. These losses are even larger during off-design conditions (Li et al., 1985). When compared to the other cycle components, the condenser in steam power cycle is subjected to higher energy loss. The pressure in the condenser determines the quantity of latent heat that is to be removed for the vapor to become condensed. The steam condenser cooling section weakens under partial load conditions and the resultant increase in vapor tends to overload the vent system at the same time as the vent system capacity is reduced at lower condenser pressures.

Dejfors et al. (1997) investigated thermodynamic advantages of utilizing ammonia-water mixtures in small direct-fired biomass fueled cogeneration plants. In the conventional condensing power application, the cycle utilizing ammonia water reaches higher power generation than the conventional Rankine steam cycle. Modifications in the cycle configuration with respect to less energy and exergy loss may lead to further improvement in power output of ammonia water cycle. Kalina cycle shows better performance at different load condition. During partial load, the performance of Rankine cycle further reduces due to variation of steam quality at the turbine exhaust. It leads to higher energy loss and reduction of LP turbine internal efficiency. In Kalina cycle, the quality of turbine exhaust is always superior, adjusting the composition will maintain proper quality of steam at the exit, and it reduces the component

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irreversibility, hence more power output. Kalina proposed a novel bottoming cycle for use in combined cycle system using an ammonia-water mixture as a working fluid. The multi component working fluid with variable boiling and condensing temperature provides less exergy loss in the evaporator and condenser. Due to that, the Kalina cycle is more efficient than the Rankine cycle especially when working with finite heat sources (Dejfors et al. 1997; Mlcak, 1996). Using ammonia-water mixture throughout the cycle is another way to improve the performance of the cycle.

The results of Dejfors et al. (1997) proved the same. Normally, using ammonia-water mixture at more than (400 °C) is not advisable, because at higher temperature NH_3 becomes unstable which leads to nitride corrosion.

2. Proposed cycle configuration and its integrated approach

The literature often suggested that combining two or more thermal cycles within a single power plant is more beneficial than operating in a single cycle alone. Two different Kalina cycle configurations like distillation condensation subsystem (Marston, 1990) and modified Kalina cycle system for geothermal resources-KCS 34 (Mlcak et al,2002) are analyzed for better performance match with the topping cycle (Rankine cycle). *Figure 1* depicts the proposed configuration of RKC cycle. RKC cycle represents the two-fluid cycles, where two cycles amalgamated in series.

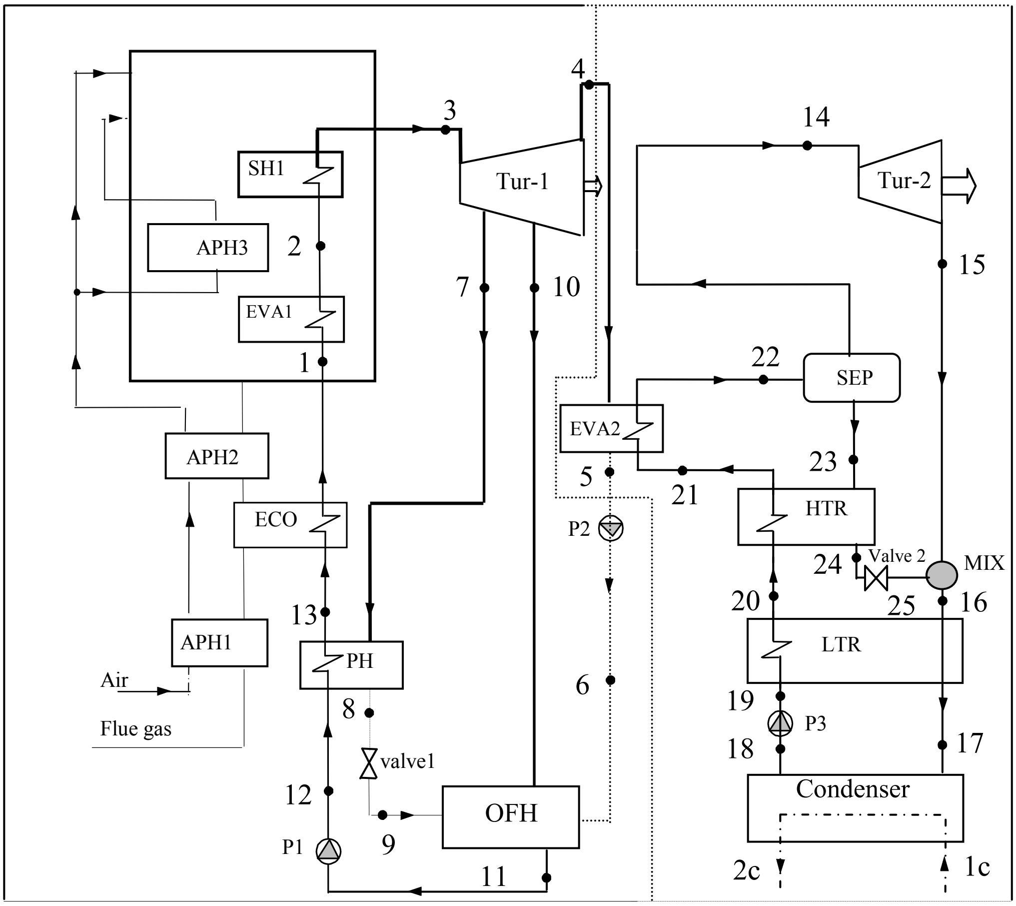

Figure 1. Scheme of proposed Rankine- Kalina combined cycle.

In all cases, the intention is to increase the cycle efficiency over that of a single cycle. A combined cycle with a different working medium is more interesting because the advantages can complement one another. The topping cycle identified in *Figure 1* is part of an 82.2 MW biomass fueled condensing power Rankine cycle. In the topping cycle (Rankine) the steam from the superheater (3) is partially expanded in the turbine and exhaust from the turbine (4) is sent to the bottoming cycle (Kalina) for further processing. In the open feed water heater, the saturated liquid from the preheater (9) is mixed with the saturated liquid from the evaporator (6). The resultant mixture is heated by bleed steam from the turbine (10). In bottoming cycle, the working fluid is in liquid phase before entering the evaporator (21). After the evaporator (22), the ammonia-water mixture splits into two streams (14, 23). The vapor (14) from the separator is expanded through the turbine. The liquid (23) gives off its heat to the incoming saturated liquid from the condenser, further throttled to the turbine exit pressure and finally it is mixed with stream (15) from the turbine exit.

3. Strategy of optimization

The first step in optimization is to transform the physical situation into a mathematical model, by identifying the number and type of variables, objective function and the constraints imposed on the system.

Figure 2 shows a T- s diagram for the RKC cycle. The state of the working fluid is identified by the same numbers as those of the schematic diagram in *Figure.1*. In the separator the ammonia water mixture is separated into liquid and vapor with different fractions of ammonia represented by thin lines (14-23 and 14-22) as shown in *Figure. 2.*

For the present case, efficiency of the cycle is considered as the objective function to optimize. The efficiency of RKC cycle depends on the following parameters:

1. Bleed steam extraction pressure (Topping cycle).
2. Fraction of ammonia-water mixture at separator inlet
3. Turbine inlet pressure (Bottoming cycle).

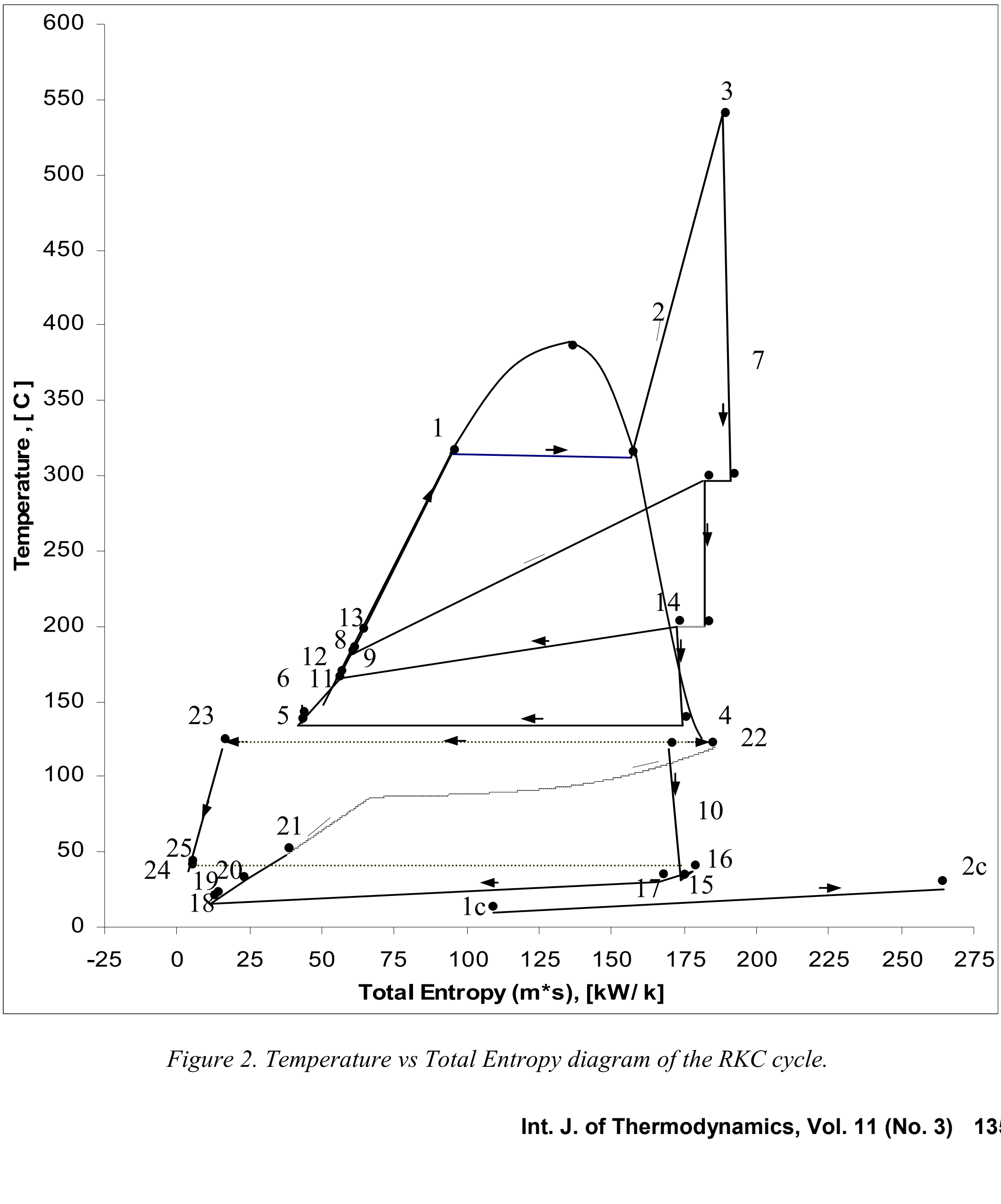

Figure 2. Temperature vs Total Entropy diagram of the RKC cycle.

The objective function to optimize is,

$$\eta_{opt} = \sum_{i=1}^{n} \eta_i - \sum_{i=1}^{n-1} \eta_i \eta_{i+1} \quad (1)$$

The variable under consideration for topping cycle is turbine extraction pressure and steam turbine outlet pressure. In bottoming cycle the optimization variables are ammonia mass fraction at the separator inlet and steam turbine inlet pressure. Checks placed throughout the program ensure that approach point, pinch point and quality of steam constraints are not violated.

To make the system optimization meaningful, it is necessary to maintain proper quality of mixture at the turbine exhaust of topping as well as bottoming cycle and appropriate pinch point and approach point must be maintained in the heat exchangers.

$$X_4 < 1 \qquad X_{14} \geq 0.90 \quad (2)$$

Using superheated steam in the bottoming cycle requires an additional super heater moreover there no benefit is obtained by using the superheated steam in bottoming cycle, therefore in this study, utilization of superheated steam is avoided in bottoming cycle.

4. About Monte Carlo method

Monte Carlo (MC) methods are stochastic techniques that use a random number generator to generate random numbers. It is a highly efficient numerical method capable of solving the most complex application (Bauer, 1958). The best solution depends on the trueness of random number. Several test points are created at random, the finest feasible of these considered is the minimum for that iteration, the search domain is reduced around the selected point, and the random trial begins again.

5. About the software program

The complete program has been written in 'C++'. For the water and ammonia-water mixture properties that are required for optimization, a separate software code was also developed using a 'C++' program by making use of the equations in the literature (Wagner et al, 1997) and the thermodynamic properties of ammonia-water mixtures were obtained by using a library of subroutines developed by Goswami et al. (1999). The software includes five different modules, taking care of steam properties, ammonia-water mixture properties, random number generator for Monte Carlo algorithm, energy analysis and finally exergy analysis. Checks placed throughout the program ensure that approach point, pinch point and quality of steam constraints were not violated.

6. Input data and assumptions

All the analyses were performed for the fuel input corresponding to 82.2 MW (Dejfors, and Svedberg, 1999). The composition of the biomass fuel is x_c=0.2499, x_{N2} =0.0020, x_{H2} =0.0304, x_{O2}=0.1980, x_{ash} =0.0098, x_{H2O}=0.5100, and LHV of the fuel is 8.43 MJ/kg and fuel rate is 9.75 kg/sec. The following assumptions were made in the cycle design.

1. Quality of steam at the turbine exit for topping and bottoming cycle should not fall below 0.90.
2. Mechanical and generator efficiency is 0.98.
3. Isentropic efficiency of the turbine 0.88.
4. Isentropic efficiency of the pump 0.80.
5. Pressure drop and heat loss in pipe lines are neglected.

7. Energy analysis of the cycle

All components associated with the cycle are steady flow devices, and thus, all processes that make up the cycle can be analyzed as steady flow processes. The kinetic and potential energy changes of the steam are usually small relative to the work and heat transfer terms and, therefore, usually neglected. In the case of the proposed Rankine-Kalina combined cycle, the heat lost by the topping cycle is absorbed in the bottoming cycle (Fig. 3). The overall cycle efficiency is the ratio of total work output to the heat input.

$$\eta = \frac{W_1 + W_2}{Q_1} \quad (3)$$

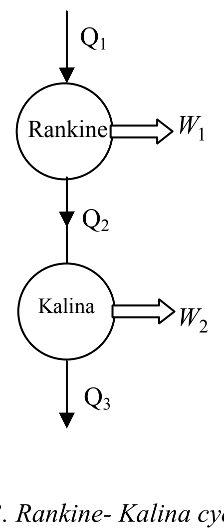

Figure 3. Rankine- Kalina cycle coupled in series.

The net cycle efficiency of Rankine cycle can be written as,

$$\eta_1 = \frac{W_1}{Q_1} \;\; or \;\; 1 - \frac{Q_2}{Q_1} \quad (4)$$

Similarly the net cycle efficiency of Kalina cycle can be written as,

$$\eta_2 = \frac{W_2}{Q_2} \ or \ 1 - \frac{Q_3}{Q_2} \quad (5)$$

Equation 3 can also be written as,

$$\eta = 1 - \frac{Q_3}{Q_1} = 1 \frac{Q_2 (1-\eta_2)}{Q_1}$$

$$\eta = 1 - \frac{Q_3}{Q_1} = 1 \frac{Q_1 \ (1-\eta_1)(1-\eta_2)}{Q_1}$$

or

$$\eta = \eta_1 + \eta_2 - \eta_1 \times \eta_2 \quad (6)$$

The cycle efficiency of topping and bottoming cycle can be written in terms of cycle parameters indicated in *Figure 1* as given below,

$$\eta_1 = \frac{(m_3 \times h_3) - (m_7 \times h_7) - (m_{10} \times h_{10}) - (m_4 \times h_4) - Wp_1 - Wp_2}{m_3 \times (h_3 - h_{13})} \quad (7)$$

$$W_{p1} = m_{12}(h_{12} - h_{11}) \,\&\, W_{p2} = m_5 (h_6 - h_5) \quad (8)$$

Where W_{P1} and W_{P2} are the pump work

$$\eta_2 = \frac{m_{14}(h_{14} - h_{15}) - W_{P3}}{m_{21} \times (h_{22} - h_{21})} \quad (9)$$

$$W_{p3} = m_{18}(h_{19} - h_{18}) \quad (10)$$

Where W_{p3} is the pump work corresponding to the bottoming cycle.

8. Exergy analysis of the cycle

Exergy is the maximum theoretical useful work (or maximum reversible work) obtained as a system interacts with an equilibrium state. Exergy analysis provides accurate information of the actual inefficiency in the system and the true location of these inefficiencies.

Exergy method shows the designer how the performance of the system departs from the ideal limit, to what extent each component contributes to this departure, and what can be done to design a better less irreversible system (Rosen, 1999).

For all exergy analysis calculations, the reference temperature is taken to be 15 °C, and the reference pressure is 1.01325 bar. The total exergy of a system becomes the summation of physical exergy and chemical exergy. The general physical exergy balance equation is given by

$$E_{ph} = (h - h_0) - T_0 (s - s_0) \quad (11)$$

In Ammonia Water mixture, the concentration of the components varies from one state to another, thus changing the chemical exergy as well as the total exergy of the working fluid. To calculate the chemical exergy of a component in the mixture the following expression is used:

$$E_{ch} = \left[\frac{e^0{}_{ch,NH_3}}{M_{NH_3}}\right] y_i + \left[\frac{e^0{}_{ch,H_2O}}{M_{H_20}}\right] (1 - y_i) \quad (12)$$

Where, $e^0{}_{ch,NH_3}$ and $e^0{}_{ch,H_2O}$ are chemical exergies of Ammonia and water. The standard chemical exergy of ammonia and water are taken from Ahrendts (1980). The chemical exergy term vanishes during irreversibility calculation.

The second law efficiency, ε, for the net power production is written as,

$$\varepsilon = \frac{Total\ exergyout\ in\ product}{Exergy\ \ in} \quad (13)$$

9. Results and discussion

Analyses were performed at different steam turbine outlet conditions and ammonia mass fractions at the separator inlet.

It is found that efficiency is best at a steam turbine exit pressure and temperature of 3 bar and 133.5 °C and the cycle configuration corresponding with the optimum parameter is depicted in TABLE I.

In bottoming cycle ammonia, mass fraction at the inlet to the evaporator and the turbine inlet pressure varied continuously to obtain the maximum power output. Optimum fraction of ammonia water mixture was found to be 0.89.

Further increases in fraction of ammonia-water mixture leads to a) decrease in mass flow rate of ammonia water mixture at the inlet separator inlet, b) decrease in mass flow rate of ammonia liquid at the inlet to the HTR, and c) decrease in work output. The variation of mass flow rate at different fractions of ammonia-water mixture are shown in *Figure. 4.* Reducing the fraction of ammonia water mixture from the optimum value leads to increase in mass flow rate of ammonia-water mixture at the separator inlet. Though mass flow rate increases, the plant output does not show much variation. The reason is increasing mass flow rate increased the quantity of work required for the pump which alleviates the benefit.

TABLE I. RESULTS FOR RKC CYCLE.

Node	P (bar)	T (°C)	y	h (kJ / kg)	m(kg/sec)	S(kJ/kg K)	Exergy Rate (kW)
1	104	313.8	0	1424	28	3.386	12601.8
2	102	312.4	0	2721	28	5.602	31039.3
3	100	540	0	3476	28	6.725	43119.0
4	3	133.5	0	2720	25	6.979	17769.5
5	3	133.5	0	561.4	25	1.672	2033.4
6	6.9	133.5	0	562.2	25	1.672	2053.4
7	17.1	296	0	3023	1.4	6.834	1477.8
8	17.1	180	0	763.6	1.4	2.139	208.6
9	6.9	164.5	0	763.6	1.4	2.144	206.5
10	6.9	199.5	0	2844	1.6	6.892	1375.8
11	6.9	164.5	0	702.65	28	2.005	3545.8
12	105	167.1	0	716.45	28	2.011	3883.8
13	105	192.3	0	822.1	28	2.244	4962.2
14	41.70	118.5	0.9728	1450.69	40.13	4.233	793276.3
15	6.917	31.61	0.9728	1227.23	40.13	4.3341	783139.8
16	6.917	36.71	0.89	961.47	50.51	3.52703	901210.5
17	6.698	30.33	0.89	887.62	50.51	3.29694	900829.0
18	6.487	15.00	0.89	-9.9476	50.51	0.227057	900171.8
19	43.33	15.98	0.89	-3.069	50.51	0.231817	900450.0
20	42.93	31.71	0.89	70.77	50.51	0.480721	900557.1
21	42.52	47.53	0.89	147.06	50.51	0.724862	900857.2
22	41.70	118.5	0.89	1215.3	50.51	3.6626	912058.4
23	41.70	118.5	0.5696	305.0	10.38	1.5056	118575.9
24	41.30	36.71	0.5696	-65.30	10.38	0.3867	118078.7
25	6.917	37.30	0.5696	-65.30	10.38	0.4007	118036.8
1c	1.03125	10	0	41.99	720.84	0.151	135.7
2c	1.01325	25	0	104.9	720.84	0.3673	557.6

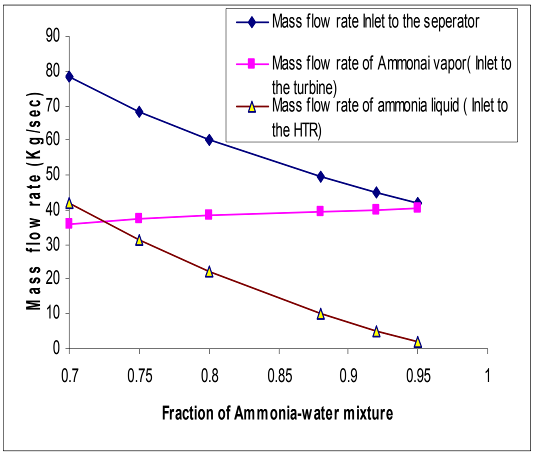

Figure 4. Variation of mass flow rate for different fractions of ammonia-water mixture.

The net power output of RKC cycle is 1.4 MW more than the power output of the condensing Rankine steam cycle configuration reported by Dejfors et. al (1997). The first law efficiency of RKC cycle is 1.43% more than condensing Rankine steam cycle. RKC cycle is having less energy loss in the condenser and LP exergy loss due to thermodynamic irreversibility in each component is calculated for the specified dead state. The exergy output depends on the degree of irreversibility of the cycle [Nag and Gupta, 1998].

The value of fuel exergy is 105.98 MW (Dejfors and Svedberg, 1999) which was obtained from the equation below (Szargut et al. 1988).

$$e_{ch,fuel} = \frac{\beta_1}{\beta_2}(LHV + h_{vap} x_{H2O}) + e_{ch,H2O} x_{H2O} \quad (14)$$

$$\beta_1 = 1.0412 + 0.2160(x_{H2}/x_C) + 0.0450(x_{N2}/x_C)$$

$$-0.2499(x_{O_2}/x_C)[1 + 0.7884(x_{H2}/x_C)] \quad (15)$$

$$\beta_2 = (1 - 0.3035)(x_{O2}/x_C) \quad (16)$$

The heat of vaporization, h_{vap}, is 2.44 kJ/kg and), $e_{ch,H2O}$=64 kJ/kg.

The exergy destructions are graphically represented by the exergy flow diagram in *Figure.5.*

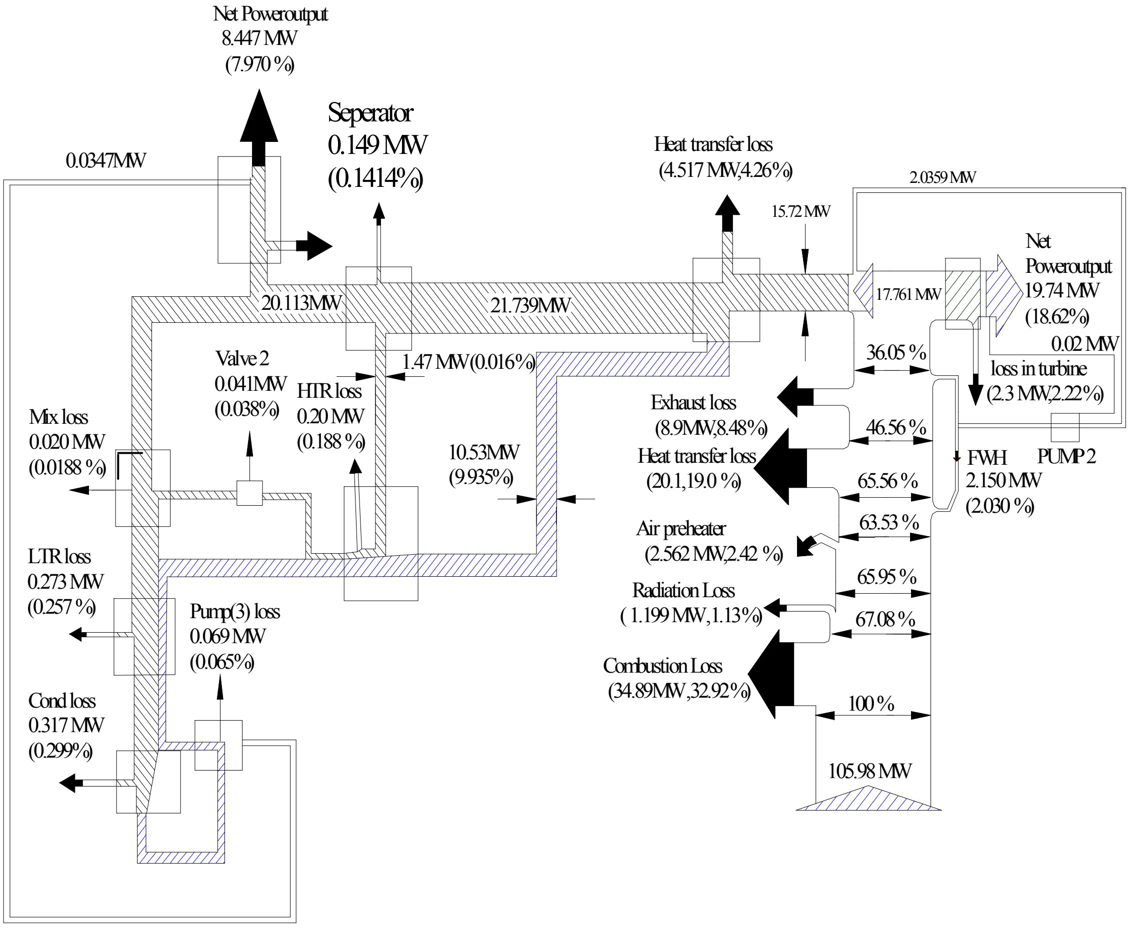

Figure 5. Exergy flow diagram for RKC cycle.

At the inlet to the condenser, ammonia water mixture is at lower temperature and, hence heat rejected in condenser is lower. In the RKC cycle, maximum output is obtained at an ammonia mass fraction of 0.89 percent and turbine inlet pressure of 41.70 bar.

Exergy flow diagram in Fig. 5 indicates that combustion isthe major thermodynamic inefficiency. In bottoming cycle the exergy losses in the evaporator is higher when compared to other cycle components in bottoming cycle and the exergy loss in the condenser is significantly less. The total exergy loss in percentage of fuel exergy in RKC cycle is around 72.70 % and it is 2.0 % less than the condensing Rankine cycle reported by Dejfors and Svedberg (1999). The thermal exergy flow diagram in Fig. 5 shows not only exergy losses but also the splitting of exergy streams and recirculation of exergy. Temperature- Enthalpy rate difference diagram is an important tool for heat exchanger analysis.

Figure 6 shows the temperature profile of heat exchange process taking place in the condenser.The temperature profile of water ammonia-mixture is highly nonlinear in nature due to variable temperature heat rejection.

Heat recuperation from the turbine exhaust fluid reduces heat rejected to the environment. This results in reduction of exergy losses in evaporator and condenser. Heat load in the condenser of a condensing Rankine cycle is

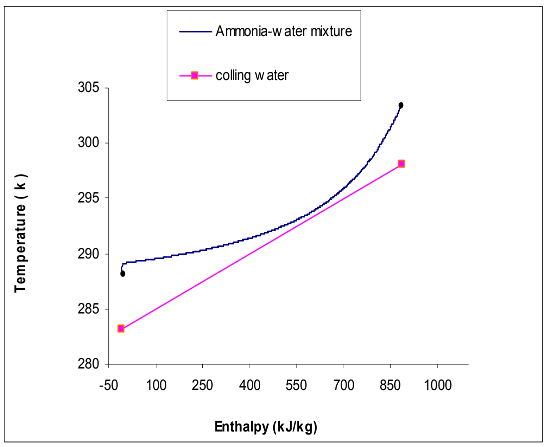

Figure 6. dT vs dh diagram for the condenser.

46.785 MW, which is 1.441 MW more than the RKC cycle. The exergy loss in the condenser is 0.299 % and is 7 % less than condensing Rankine steam cycle. It confirms that RKC cycle has less energy as well as exergy loss in the condenser. Second law efficiency of RKC cycle is around 27.22 % and is 2.0 % more than the condensing Rankine cycle configuration adopted for this study.

10. Conclusions

The current study explored the possibility of integrating two different cycles for the sake of better performance. The overall energy and exergy analysis were performed to find out the thermodynamic performance of proposed RKC cycle. The author proposed a new approach for reducing energy loss due to moisture in the turbine exhaust and losses in the condenser of Rankine steam cycle power plant. The energy and exergy results shows that proposed Rankine-Kalina combined cycle is more efficient than Rankine steam cycle operating on a condensing mode.

In the topping cycle all the parameters that we used for this analysis pertain to one of the direct-fired 82.2 MW_{fuel} biomass fueled Rankine cycle power plants in Sweden. Addition of Ammonia-Water cycle as a bottoming cycle to the real direct-fired biomass plant provides the following benefits.

1. The condenser pressure in Rankine steam cycle always operates under vacuum, whereas in RKC cycle condenser pressure is more than atmospheric pressure. Due to that an air removal system and dearation are not required for RKC cycle. In RKC, cycle condenser pressure depends on cooling water inlet temperature unlike Rankine cycle power plant in which it depends on cooling water outlet temperature. Energy loss in the condenser is less when compared to energy loss in the Rankine cycle.

2. Since the specific volume of steam at the turbine exhaust of RKC cycle is lower than that of the Rankine cycle, the turbine system and exhaust is very small.

The cost of electricity for RKC cycle may be substantially lower only if the cost associated with the additional components in the RKC are not excessive compared to that for a condensing Rankine steam cycle.

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Nomenclature

E	Exergy flow rate [kW]
e	Specific exergy [kJ/kg]
h	Enthalpy [kJ/kg]
s	Entropy [kJ/kg K]
W_1	Net power output topping cycle [kW]
W_2	Net power output bottoming cycle [kW]
Q_1	Heat added in topping cycle [kW]
Q_2	Heat added in bottoming cycle [kW]
Q_3	Heat rejected from the bottoming cycle [kW]
p	Pressure [bar]
t	Temperature [°C]
m	Mass flow rate [kg/sec]
X	Quality of steam at the turbine exhaust
n	Number of cycle
y	Ammonia mass fraction in the solution
M	Molecular weight

Subscripts

ph	Physical exergy
ch	Chemical exergy

Greeks

η_1	Topping cycle efficiency
η_2	Bottoming cycle efficiency

Abbreviation

RKC	Rankine- Kalina combined cycle
LTR	Low temperature recuperater
HTR	High temperature recuperater
FWH	Feed water heater

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