NEW KALINA CYCLE FOR POWER GENERATION

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Abstract

With the utilisation of binary mixture as working fluid in a power generation cycle, higher heat conversion rate will be achieved in the boiler which results in more efficiency. In this paper two kalina cycle designs using binary mixtures as working fluid were compared, the cycle efficiency as a function of turbine inlet and separator outlet were calculated for the kalina cycles. The cycles property values at each points are obtained from matlab. The optimized design were identified at various turbine inlet conditions resulting in energy conservation.

Keywords: New thermodynamic cycle, Ammonia-Water mixture, MATLAB

I. INTRODUCTION

Power Cycles is a thermodynamic cycle which converts thermal energy into power and stores power in a generator. In power production two different aspects are in practice. One with direct power production and the other is indirect way of producing power. In the direct power production method the conversion method that exhibits direct power production include processes in which the working fluid is heated by the source is the only intermediary. Examples are solar steam expansion, geothermal steam expansion and solar chimney. In the second method, the fluid is heated by the source is an intermediary transfer fluid, which subsequently transfer energy to the working fluid that produces power. The power production can occur simultaneously with the heat extraction from the source, or the charged transfer fluid can be stores for later generation. Examples are Stirling cycle, Rankine cycle, and Kalina cycle.

Kalina cycle is a thermodynamic cycle, produces power utilizing binary mixture as working component. A well known Kalina Cycle produces O.M. Ibrahim and S.A. Klein concluded the kalina Cycle produces about 80% of the maximum power at a very high thermal capacitance ratio [1]. T.Heppenstall identified Kalina as a bottoming cycle shows better performance [2]. C. Dejfors, E. Thorin and G. Svedberg proposed the ammonia-water mixture reaches high power generation than the single component [3]. P.K. Nag concluded by increasing the turbine inlet temperature and the separator temperature the cycle efficiency increases [4]. While calculating the performance of the power cycle E. Thorin, C. Dejfors, and G. Svedberg identified the correlations for the thermodynamic properties (temperature, pressure, volume, enthalpy and entropy) of $NH_3 - H_2O$ mixture play an important role [5]. In Kalina cycle the ratio of exergy loss with the net generated power was less compared with the Rankine cycle as proposed by Thongcai srinophakun, sangapong Laowithavangkul, Masaru Ishida[6]. A comparison between Kalina cycle and ORC were made by Roland Dippio and concluded among the binary plants Kalina cycle generates 30% to 50% more power for a given heat source. With the Kalina cycle as a bottoming cycle for a cogeneration plant Jose A. Borgert and Jose A. Velasquez proposed the exhaust gas temperature was reduced from 427 K to 350 K which reduces environmental impact [7]. Mark Mirolli concluded the DCSS technology is a key component for the high efficiency of a Kalina cycle plant for waste heat recovery power plant applications [8]. VasileMinea conceived the Kalina cycle may produce power in the future especially with industrial waste heat and biomass [9].

A Kalina cycle is a power generation cycle which uses non-azeotropic (ammonia-water) mixture as working fluid for increased thermal efficiency. With the utilization of a non-azeotropic mixture the change in temperature during boiling and condensation of the mixture will result. Due to this a closer match between heat source and the working fluid is achieved. With the binary mixture, boiling exists at a lower temperature than a single component and with the same amount of fuel supply more amount of steam will be extracted from the working component. In a steam cycle the condensation at high temperature is impossible.

Whereas in Kalina cycle the condensation at high temperature and low pressure is achieved by the incorporation of separator. With the utilization of binary mixture the following improvements can be achieved.

(i) Reduction in fuel proportion with the gains in efficiency.

(ii) For condensation low pressure is sufficient.

(iii) For spinning the turbine high pressure is obtained.

Why Kalina Cycle?

- In Rankine cycle more than half of the heat transfer occurs during the boiling process which is considered as constant temperature boiling process.
- In ORC utilizing isopentene as working fluid, isopentene boils at a constant temperature under a given pressure.
- In Organic Rankine cycle the theoretical efficiency and cost/production ratio are less.

Present work calculates the efficiency of the existing real time cycle (Hawaii Plant) using MATLAB. The practical efficiency calculation was compared with the simulated value and happened to find a percentage error of less than 5%. In addition to which a high temperature plant was considered and the same practical and simulated efficiencies were compared.

In addition, a new model is proposed and the efficiency calculation were made.

II. CORRELATIONS FOR CALCULATING THEMODYNAMIC PROPERTIES

For determining the performance of the power cycle correlations the thermodynamic properties were calculated. In this work the correlations proposed by Ziegler and Trepp, Patek and Klomfar and Soleimani were utilised and developed in MATLAB. The Gibbs free energy for both liquid and vapor phases were used for determining enthalpy, entropy and volume. The correlations were coded in MATLAB and executed.

Hawaii Plant

The largest "waste heat" resource available in Hawaii for kalina cycle application is the heat stores in the tropical region surrounding the state.

The 10000 kW hawaii plant has inlet temperature of 26°C.

The process uses a binary working fluid of ammonia and water with distillation / condensation subsystem which enables different working fluid mixtures at different stages of the cycle with heat recuperative stages for increased efficiency.

The DCSS provides essential to the existence in establishing the high ammonia-water concentration for the heat acquisition stage and a low ammonia-water concentration at the condensation stage.

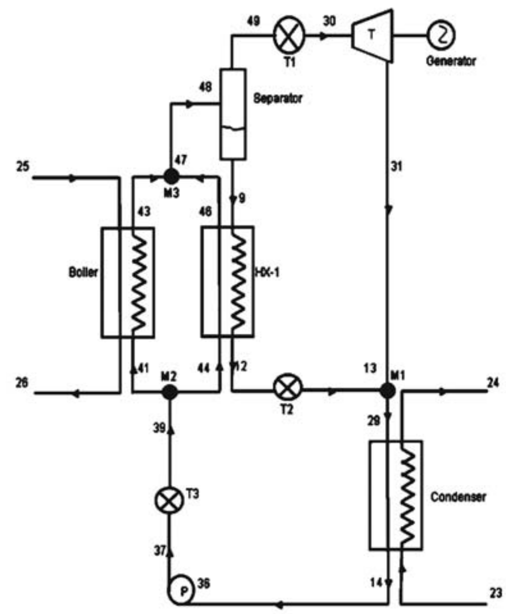

Fig. 1. Hawaii Kalina Cycle Power Plant

Table 1 Plant Performance Summary

	Plant	Simulation
Heat	In 513897.50 kW	513897.50 kW
Heat Rejected	503729.64kW	503729.64kW
Power Turbine	10526.32 kW	10048.34 kW
Gross Generator Power	10105.26 kW	9646.40 kW
Power Pumps	382.25 kW	427.96 kW
Net Output	10144.07 kW	10476.3 kW
Thermal Efficiency	1.97%	2.02%
Plant Efficiency	1.54%	1.46%

Table 2. Comparative state point values of Hawaii plant with the simulated results

Point	X Plant	X Simulation	T Plant	T Simulation	P Plant	P Sim.	H Plant	H Sim.	Mass Plant	Mass Sim.
9	0.68	0.69	22.95	23	5.887	5.919	–103	–97.57	600.41	647.35
10	0.68	0.69	22.79	22.83	5.852	5.88	–103	–98.31	600.41	647.35
12	0.68	0.69	17.88	17.33	5.576	5.86	–127.78	–118.79	600.41	647.35
13	0.68	0.69	14.64	17.16	4.42	4.5	–127.78	–118.79	600.41	647.35
14	0.802	0.802	6.78	7	4.353	4.39	–114.65	–111.15	967	1013.9
23	Brine	Brine	4	4	–	–	16.75	15.75	16862.95	16862.95
24	Brine	Brine	11.13	11.13	–	–	16.62	16.62	16862.95	16862.95
25	Brine	Brine	26	26	–	–	103.9	103.9	16773.04	16773.04
26	Brine	Brine	18.33	18.33	–	–	73.27	73.27	16773.04	16773.04
29	0.802	0.802	14.45	14.13	4.42	4.54	406.27	396.68	967	1013.9
30	0.9998	0.9998	22.88	23	5.861	5.86	1309.68	1312. 1	366.59	366.59
31	0.9998	0.9998	8.11	10	4.444	4.44	1280.97	1284.7	366.59	366.59
32	0.9998	0.9998	8.03	9.9	4.42	4.39	1280.97	1284.7	366.59	366.59
33	0.802	0.802	14.45	14.13	4.42	4.39	406.27	388.65	967	1013.9
36	0.802	0.802	6.78	7	4.353	4.39	–114.65	–111.15	967	1013.9
37	0.802	0.802	6.82	7	6.466	5.91	–114.28	–110.63	967	1013.9
38	0.802	0.802	6.82	7	6.466	5.91	–114.28	–110.63	967	1013.9
39	0.802	0.802	6.83	7.007	6.1	5.83	–114.28	–110.63	967	1013.9
40	0.802	0.802	6.83	7.007	6.1	5.83	–114.28	–110.63	967	1013.9
41	0.802	0.802	6.83	7.007	6.059	5.83	–114.28	–110.63	934.3	983.95
43	0.802	0.802	23.22	23	5.92	5.77	435.86	411.51	934.3	983.95
44	0.802	0.802	6.83	7.007	6.059	5.83	–114.28	–110.63	32.86	29.98
46	0.802	0.802	20.56	20	5.92	5.77	325.77	409.61	32.86	29.98
47	0.802	0.802	22.95	20	5.887	5.77	432.11	409.61	967	1013.9
48	0.802	0.802	22.95	23	5.887	5.91	432.11	478.42	967	1013.9
49	0.9998	0.9998	22.95	23	5.887	5.91	1309.68	1311.5	366.59	366.59
50	0.9998	0.9998	22.93	22.77	5.877	5.86	1309.68	1310.6	366.59	366.59
51	0.9998	0.9998	22.91	22.54	5.872	5.80	1309.68	1310.6	366.59	366.59
52	0.9998	0.9998	22.88	23	5.861	5.86	1309.68	1310.6	366.59	366.59

III. THERMODYNAMIC CYCLE

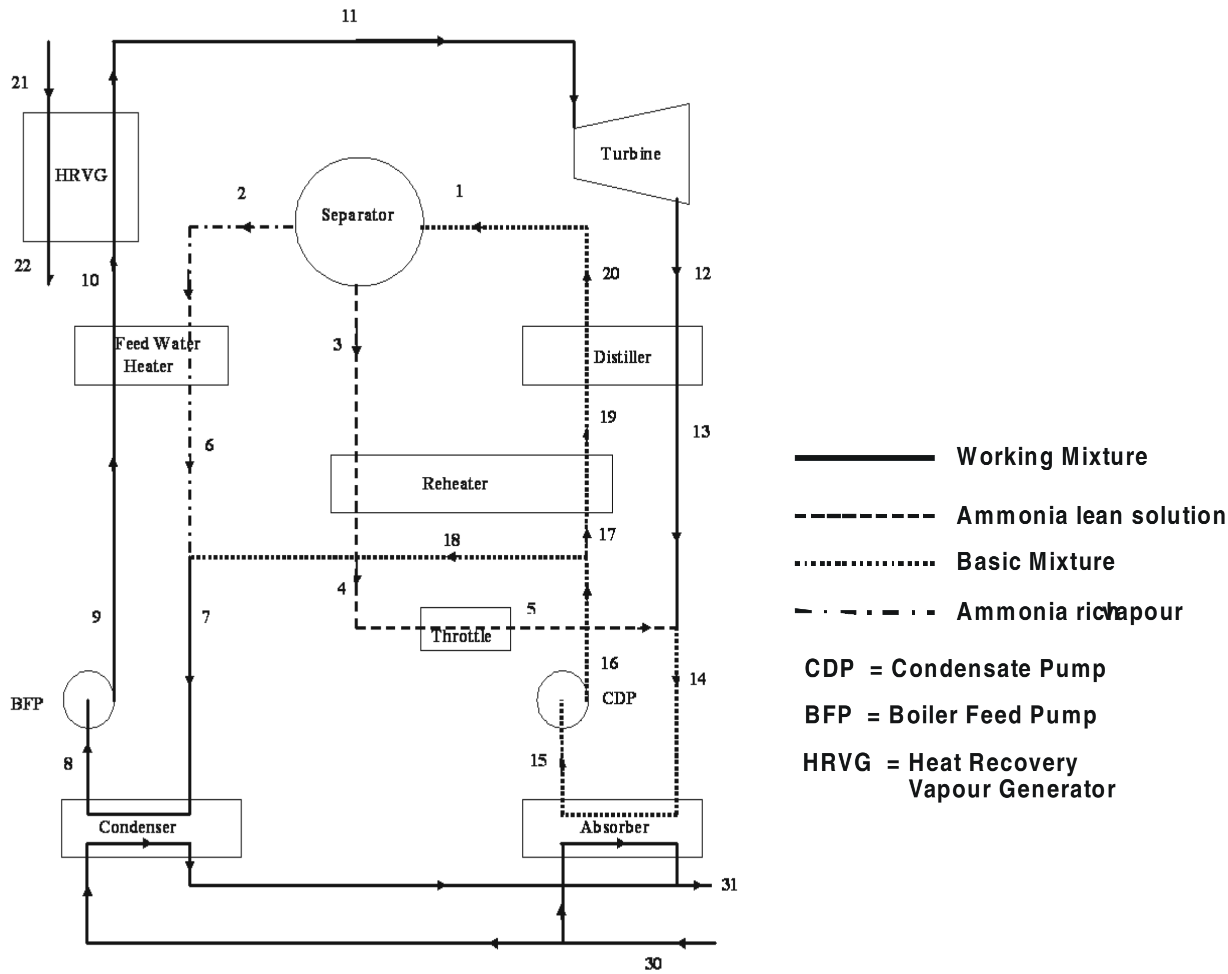

Fig. 2. Schematic diagram of Kalina Cycle

The working fluid ammonia-water mixture is vaporized and superheated in the HRVG before expansion through the turbine. The relatively high concentration of ammonia in the working fluid is called the working mixture composition. The superheated vapor (11) is then expanded in the turbine (12) to transform its energy into useful form. The generated spent stream is then cooled in a distiller (13) and diluted (5) with a weak solution resulting in basic mixture composition (14). The basic mixture composition raises the condensing temperature and condensed in the absorber. A stream with a high concentration of ammonia, like the turbine outlet stream, cannot be condensed by cooling water of a normal temperature, since the high ammonia concentration would result in a very low condensation temperature at the pressure level in the condenser. A pump (15) increases the pressure of the basic mixture condensate and the stream is split: one (17) of the resulting streams is sent to the separator via the reheater and the other stream (18) is mixed with the ammonia-rich vapor from the separator to restore the working mixture concentration. One of the resulting streams (1) is sent is sent to the separator via the reheater recovering heat from the turbine outlet stream. The flash tank separator produces one stream of ammonia-lean saturated liquid (3) and one stream of ammonia-rich saturated vapor (2). The ammonia-lean liquid stream gives up heat in the reheater , then throttled (5)and absorbs the working mixture stream (14) from the turbine before condensation in the low-pressure condenser. The other stream (18) is mixed with the ammonia-rich vapor (6) from the separator to restore the working mixture concentration (7). Then it is condensed in a condenser (8), pressurized in a boiler feed pump (9) and sent into the

HRVG where it is superheated by the source. The real focus of any power cycle is to increase the cycle efficiency by reducing the structural losses. Reduction in structural losses (Figure 2) results in increased actual work. In a single component working fluid cycle as the boiling and condensation temperature is constant, therefore the working fluid curve is not paralleled with the heat source curve.

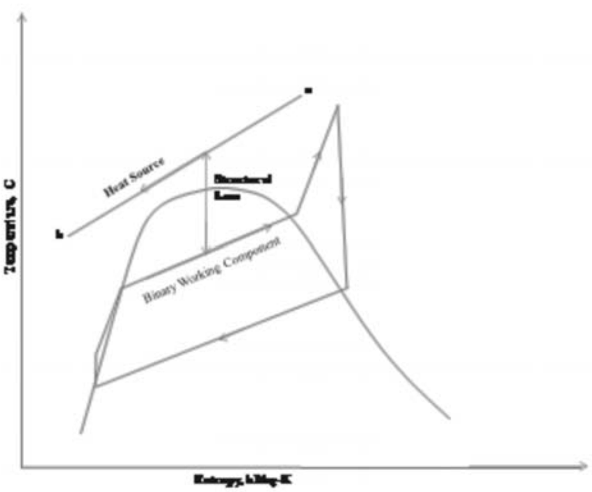

Fig. 3. T-S plot of Kalina Cycle with heat source

IV. THERMODYNAMIC ANALYSIS

To carry out thermodynamic analysis Energy balance, Mass balance and Material balance were to be find out. The first step in calculating the performance of the cycle is to perform mass balance and energy balance. The equations for all the components were made out.

SEPARATOR:

For the input parameters, pressure, concentration and temperature using MATLAB the liquid concentration X_1 and Vapor concentration Xv was calculated.

TURBINE:

$$h_{11} = h_{12} - \eta_T (h_{12} - h_{11s})$$

where, $s_{12} = s_{11s}$

PUMP:

$$h_{16} = h_{15} - (h_{15} - h_{16}/\eta_p \text{ Where, } s_{15} = s_{16s}$$

$$h_9 = h_8 (h_8 - h_{9s})/\eta_p \text{ Where, } s_8 = s_{9s}$$

For pump and turbine the efficiencies

were assumed as $\eta_T = 0.9$ $\eta_p = 0.6$

REHEATER:

$$m_3 (h_3 - h_4) = m_{19} (h_{19} - h_{17})$$

In reheater, distiller, and feed water heater the Terminal Temperature Difference was assumed as 5°C to 10°C

MIXER:

$$m_5 + h_5 + m_{13} h_{13} = m_{14} h_{14}$$

$$m_6 h_6 + m_{18} h_{18} = m_{17} m_{17}$$

With these equations the enthalpy, Entropy and volume values for all the Points considered in the cycle was obtained using MATLAB. The thermodynamic properties evaluated

using the correlations proposed were helpful in obtaining the values at each points in a fast manner.

V. EFFICIENCY CALCULATION:

The efficiency of the cycle is the ratio of the output with the input. The output is considered as the turbine work and pumps work. The input is consided at the HRVG.

TURBINE WORK:

$$W_t = m_{11} (h_{11} - h_{12})$$

BOILER PUMP WORK:

$$P_{w1} = v_8 (p_8 - p_9) = m_8 \times R (T_8 - T_9)$$

Table 2 Input Parameters

Point	P, bar	T°C Existing	T°C	X Existing	X obtainted	H, kJ/kg Existing	H, kJ/kg obtained	S, kJ/kg – K Existing	Z, kJ/kg – K obtained	M, Kg Existing	M, Kg obtained
1	5.539	70	70	0.448	0.448	312	376	1.49	1.33	2.94	3.23
2	5.539	70	70	0.9685	0.979	1482	1450.95	5.2	4.75	0.48	0.47
3	5.539	70	70	0.3459	0.3568	82	93	0.76	0.69	2.46	2.75
4	5.539	25.06	25.05	0.3459	0.3568	– 117	– 114.8	0.14	0.0425	2.46	2.75
5	2	25.17	25	0.3459	0.3568	– 117	– 114.8	0.14	0.04	2.46	2.75
6	5.539	27.36	27	0.9685	0.979	1316	1283.88	4.7	4.22	0.48	0.47
7	5.539	42.92	39	0.7	0.7	557	603	2.3	1.84	1.00	1.00
8	5.539	20	20	0.7	0.7	– 107	– 107	0.10	0.007	1.00	1.00
9	100	23.56	22	0.7	0.7	– 80	– 87.6	0.13	0.02	1.00	1.00
10	100	39.09	42	0.7	0.7	0.007	0.409	0.40	0.32	1.00	1.00
11	100	500	500	0.7	0.7	2798	2799	6.27	6.208	1.00	1.00
12	2	118.87	113	0.7	0.7	1897	1888	6.52	5.45	1.00	1.00
13	2	57.47	59	0.7	0.7	1041	1198	4.0	3.559	1.00	1.00
14	2	40.15	39	0.448	0.7	209	234	1.27	0.97	3.46	3.75
15	2	20	20	0.448	0.448	– 154	– 156	– 0.043	– 0.068	3.46	3.75
16	5.539	20.06	20.05	0.448	0.448	– 154	– 155.9	– 0.044	– 0.068	3.46	3.75
17	5.539	20.06	20.05	0.448	0.448	– 154	– 155.9	– 0.044	– 0.068	0.51	0.52
18	5.539	20.06	20.05	0.0448	0.448	– 154	– 155.9	– 0.044	– 0.068	0.51	0.52
19	5.539	52.47	54	0.448	0.448	12	28	0.58	0.42	2.94	3.23
20	5.539	70	70	0.448	0.448	312	376	1.49	1.33	2.94	3.23

CONDENSATE PUMP WORK:

$$P_{w2} = v_{15}\,(p_{15} - p_{16}) = m_{15} \times R\,(T_{15} - T_{16})$$

HRVG:

$$Qs = m_{11}\,(h_{11} - h_{10})$$

EFFICIENCY:

$$\eta = \frac{W_t - P\,w_1 - P\,w_2}{Qs}$$

With this calculation the efficiency of the cycle was obtained.

VI. RESULTS AND DISCUSSIONS:

Table 1 gives the values of the cycle nodes chosen. The poperty values as well as the mass and energy values. The values obtained were calculated using the proposed correlations in MATLAB. The values obtained by the present result are comparibly closer with the existing results with deviations upto 2% to 5%. The important parameters which affect the cycle efficiency was assessed as turbine inlet temperature, separator temperature and turbine inlet concentration. The effeciencies at various separator temperature and turbine inlet concentration were calculated and plotted. Figure 3 shows the cycle efficiency graph as a function of separator temperature and turbine inlet concentration with the constant turbine inlet temperature. At constant turbine inlet temperature, the separator temperature decreases with the increase in the turbine inlet concentration. The cycle efficiency as a function of turbine inlet temperature and inlet concentration is shown in the figure 4 the turbine inlet temperature inlet concentration.

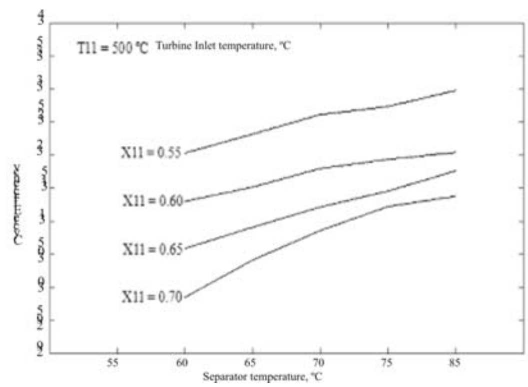

Fig. 4 Cycle efficiency as a function of separator temperature and turbine inlet Concentration

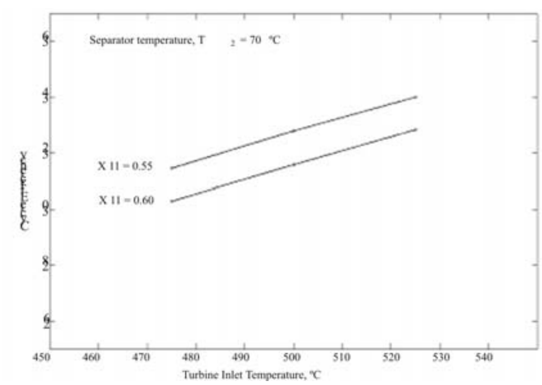

Fig. 5 Cycle efficiency as a function of Turbine Inlet temperature and turbine inlet Concentration

A. NEW KALINA CYCLE APPLICABLE FOR LOW PRESSURE APPLICATIONS

WORKING PRINCIPLE:

The cycle shown in the figure.5 is suitable for low pressure applications. The cycle shown is used for converting energy from low pressure, moderate temperature stream (external source) into usable energy by a binary mixture working component. The cycle consists of high pressure circuit and low pressure circuit. The present cycle involves eight heat exchangers, a separator, a turbine, two pumps, a mixer and a separator. In this cycle the irreversibility in the process of mixing the basic solution with the re-circulating solution was reduced by vaporizing the basic solution completely and preheating the re-circulating before mixing which reduces the irreversibility in the process of mixing and increases the efficiency of the overall process. Also the flow rate of the working solution passing through the turbine will be increased, thus an increased power output. Initially the basic solution is pumped in a pump (P1) and pressurised which then is heated in a recuperative preheater by condensed basic solution. During the first heat exchange process and during the preheated steam and condensed steam will be produced. The preheated steam at a state of saturated state is splitted into two streams, one of the two streams is passed in a heat exchanger where it is partially vaporized by the counter flow

heat source fluid stream and considered as second heat exchange process. The other stream is partially vaporized by the condensing working fluid stream . The two partially vaporized streams are combined at a state of liquid-vapor mixture. The mixture is further heated and vaporized with the cooled heat source in a fourth heat exchange process resulting in a state of saturated vapor.The working solution stream is then heated and vaporized with a first cooled heat source fluid stream in a fifth heat exchange process producing saturated vapor. The vapor is heated in heat exchanger producing superheated vapor which is then expanded in a turbine. The spent stream is used to heat the preheated stream in the recuperative preheater.

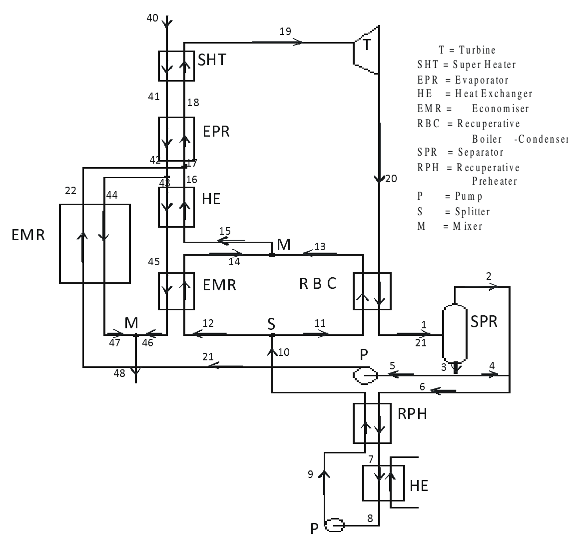

Fig. 6 New Kalina Cycle Applicable for Low Pressure Applications

Table 3 Property values of the at cycle points

Point	X	T (°C)	P, (bar)	h, (kJ/kg)	s, (kJ/kg-K)	m, (kg)
1	0.8245	77	8.25	1070	3.89	1.17
2	0.952	77	8.25	1440	4.67	0.87
3	0.385	77	8.25	127	0.77	0.29
4	0.385	77	8.25	127	0.77	0.17
5	0.385	77	8.25	127	0.77	0.12
6	0.9	77	8.25	1280	4.3	1
7	0.9	45	8.10	966	3.12	1
8	0.9	21	8.09	20.54	0.237	1
9	0.9	21	32.65	22.72	0.24	1
10	0.9	73.9	31.99	283.37	1.04	1
11	0.9	72	31.99	283.37	1.04	0.60
12	0.9	73.9	31.99	283.37	1.04	0.40
13	0.9	104	31.87	1284	4.26	0.60
14	0.9	104	31.87	1284	4.26	0.40
15	0.9	104	31.87	1284	4.26	1
16	0.9	142	31.87	1632	4.70	1
17	0.8245	142	31.87	1454	4.40	1.17
18	0.8245	160	31.74	1758	4.95	1.17
19	0.8245	183	31.70	1826	5.0	1.17
20	0.8245	109	8.49	1582	4.75	1.17
21	0.385	77	31.99	127	0.7	0.17
22	0.385	142	31.87	478.7	1.77	0.17
31	0.385	77	31.99	127	0.7	0.17
40		187	1.013	786	2.26	2.58
41		179	1.013	723.83	2.05	2.58
42		147	1.013	615.04	1.59	2.58
43		147	1.013	615.04	0.90	0.18
44		109	1.013	343	1.68	0.18
45		109	1.013	500	1.65	2.40
46		147	1.013	615.04	0.98	2.40
47		82	1.013	343	1.10	0.18
48		77	1.013	319	1.04	2.58

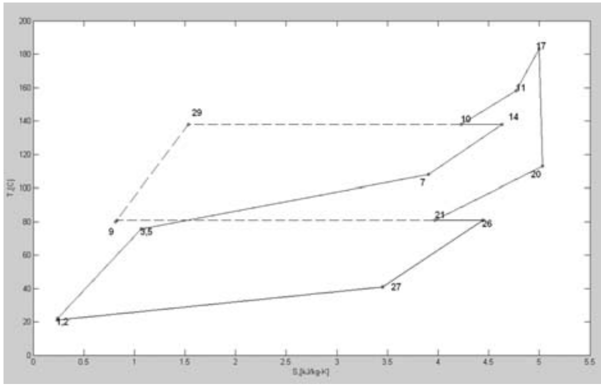

Fig. 7. T-S Plot of the proposed model

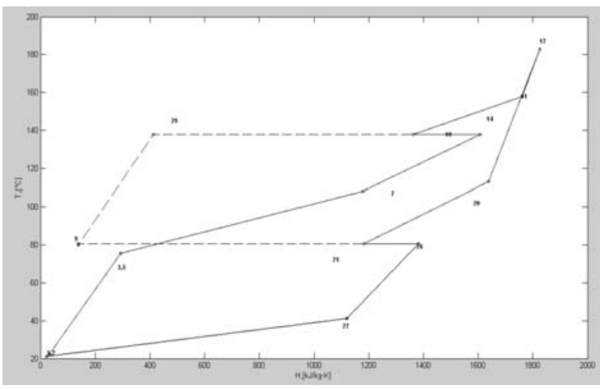

Fig. 8. T-H Plot of the proposed model

Table. 2 shows the property values at all cycle points chosen. Based on the energy and mass balance equation as calculated for the above cycle, the property values at each cycle points were calculated. The efficiency of the cycle is then calculated.

Figure 9 shows the cycle efficiency plot with the constant parameter turbine inlet temperature, variable separator temperature and turbine inlet concentration. The trends shows that the cycle efficiency decreases at increased concentration.The trends were obtained with the parameters identified in affecting the efficiency of the cycle. At constant Turbine inlet temperature and at various separator temperatures the cycle efficiency were calculated. Also the cycle efficiency were calculated by considering constant separator temperature and varied turbine inlet temperature.

VII. CONCLUSION

The main goal of this work is to extract more energy from the heat source and efficiently converted to work output. The basic cycle was considered and using MATLAB the properties were calculated as the preliminary measure. The efficiency was calculated and compared with the existing results. With the new cycle applicable for moderate temperature and low pressure applications will utilize the heat source much more efficiently as the recirculating solution is combined with the working solution which in turn increases the heat load with a high flow rate in the turbine inlet. With the utilization of multiple heat exchangers the complete vaporization in the super heater and complete condensation in the condenser is achieved.

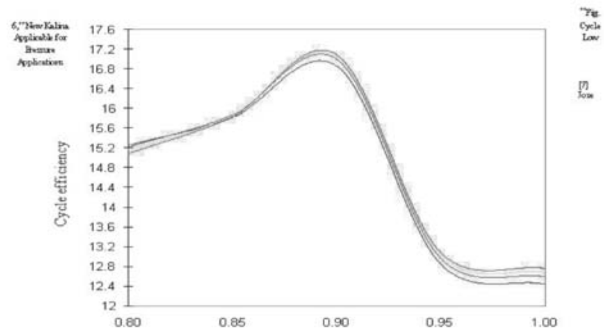

Fig. 9 New Kalina Cycle Applicable for Low Pressure Applications

Nomenclature

m	mass flow
h	specific enthalpy, kJ/kg
s	specific entropy, kJ/kg-K
v	specific volume, m^3/kmol
T	temperature, *K*
p	pressure, bar
□	cycle efficiency
W	Power
P	Pump
R	Gas Universal Constant

Subscripts

t	turbine

REFERENCES

[1] Ibrahim O.M. and Klein S.A .(1996). “Absorption Power Cycles”, Energy (1), pp. 21-27B.

[2] Heppenstall T. (1998). “Advanced Gas Turbine Cycles for Power Generation a Critical Review”, Applied Thermal Engineering (18) pp. 837-846.

[3] Dejfors C., Thorin E., and Svedberg G. [199]. “Ammonia-Water Power Cycles for Direct-Fired Cogeneration Applications” (16) pp.1675 – 1681.

[4] Nag P.K. and Gupta A.V.S.S. (1998) “Exergy Analysis of Kalina Cycle”, Applied Thermal Engineering (18) pp.427-439.

[5] Thongchai Srinophakun, Sangapong Laowithayangkul, Masaru Ishida (2001) “Simulation of Power Cycle with Energy Utilization Diagram” Energy Conversion and management” pp.1437-1456.

[6] Ronald DiPippo (2004). “Second Law assessment of binary plants generating power from low-temperature geothermal fluids”, Geothermics (33) pp. 565-586.

[7] Jose A. Borgert and Jose A.Velasquez (2004). “Exergoeconomic Optimization of a Kalina Cycle for Power Generation” Int.J.Exergy Vol.1.No.1.

[8] Mark D.Mirolli (2005). “The Kalina cycle for Cement Kiln Waste Heat Recovery Power Plants” IEEE.

[9] Vasile Minea (2007). “Using Geothermal Energy and Industrial Waste Heat for Power Generation” IEEE.

[10] Srinivas (2008). “Generalised Thermodynamic Analysis of Steam Power Cycles with ‘n’ number of feed water heaters”Int.J. of Thermodynamics (4), pp. 177-185.