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Comparative study on steam flash, organic flash and Kalina for enhanced power generation from waste heat recovery

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Abstract

A huge amount of waste hot gases from cement factories are available for power generation. The current work is focused on enhancing the power from the hot gas with proper power augmentation and selection of technology. The selection of power plant and working fluid influences the output from waste heat recovery. In the current work, three power plants are considered at low temperature heat recovery (LTHR) and five power plants are considered at intermediate temperature heat recovery (ITHR). Organic Rankine cycle (ORC), organic flash cycle (OFC), Kalina cycle system (KCS) are studied at LTHR and ITHR and steam Rankine cycle (SRC) and steam flash cycle (SFC) are studied at ITHR. R124 and ammonia are selected respectively at LTHR and ITHR organic cycles (ORC and OFC). Following a comparative study, OFC is recommended for LTHR and KCS is recommended for ITHR on the basis of maximum power generation.

Keywords: energy, efficiency, flash cycle, heat recovery, Kalina, vapor absorption

1. Introduction

Cogeneration is the most economic option to generate power from waste heat recovery without depending on the combustion of conventional fuels. A typical cement factory requires a furnace to generate high temperature gas for cement processing. After use, there is a great potential in the waste gas to generate power. The success of power generation depends on selecting the right power plant configuration and working fluid to increase heat recovery and hence power. The steam Rankine cycle (SRC) and organic Rankine cycle (ORC) are used by regular power plants. The organic flash cycle and Kalina cycle system (KCS) are selected for LTHR and compared with ORC. Similarly, the steam flash cycle (SFC), OFC and KCS are selected for ITHR and compared with regular power plants (SRC and ORC).

A steam power plant is suitable for recovering heat at intermediate and high temperature levels. Many researchers have worked on steam bottoming cycles to recover gas turbine exhaust heat in a typical combined cycle power plant [1, 2]. At high temperature, a multi pressure heat recovery steam generator delivers better results than a single pressure system [3]. Wang et al. [4] described the steam flash cycle suitable for a cement factory waste heat. The use of SFC avoids the use of multi pressure heat recovery, which is complex and costly. The flash cycle is simple, cost effective and easy to operate. It differs from the flash cycle in a geothermal power plant [5]. In a geothermal power plant, the pressurized water is flashed completely and the flashed steam is supplied to the turbine. The efficiency of this cycle is low due to the supply of saturated steam to the turbine without super heat. Pradeep Varma and Srinivas [6] reported a case study of a regular steam power plant which uses the waste heat of a cement factory.

Thermodynamic properties of steam (critical pressure, critical temperature, latent heat etc,) are well suited for a heat recovery. The steam temperature is constant in phase change therefore the temperature gap is more in the evaporator of HRSG. Steam is not suitable for LTHR. Chacartegui et al. [7] worked on alternative ORC cycles to recover heat in the place of steam, but with relatively lower source temperature, and suggested toluene and cyclohexane for higher efficiencies. Quoilin et al. [8] recommended n-butane as a working fluid for waste heat recovery working ORC after thermodynamic and economic examination. Wang et al. [9] considered the surface area of heat exchanges in ORC and suggested isobutene as the best fluid after optimization. It is possible to integrate the steam power plant with ORC, but it adds complexity and cost even though it does give a considerable boost in efficiency [10]. Sarkar and Bhattacharyya [11] rec-

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Email address: `srinivastpalli@yahoo.co.in` (G. V. Pradeep Varma) ommended ammonia as a working fluid for more power, and compact turbomachinary and n-pentane for high efficiency and compact heat exchangers.

A zeotropic mixture allows the phase change with variable temperature so that the temperature glide will match the source temperature. Heberle et al. [12] checked the feasible benefit of zeotropic mixture use over the pure substance in ORC. They did not use the flashing option in the zeotropic ORC. But it demands relatively high pressure to condense at the sink temperature. To avoid this difficulty, the hot pressurized zeotropic mixture is flashed into liquid and vapor. In a typical KCS, the dilution of working fluid with water avoids the operation of a condenser at high pressure [13]. Li and Dai [14] gave preference to KCS over CO_2 transcritical cycle after the thermo-economic evaluation to recover low temperature waste heat for power generation. Kotowicz et al. [15] show the potential of waste heat recovery to generate low cost electricity through the organic cycle module.

Examination of the literature indicates that the flash cycle is not well formulated and analyzed critically. A comparative study of this flash cycle with steam and organic fluid is also not found in the literature. In the current work three power plants (ORC, OFC and KCS) and five power plants (SRC, SFC, ORC, OFC and KCS) are compared with a view to assessing the benefits of flashing withregard to heat recovery from the cement factory. Study of cement factory processes lies outside the scope of the current work.

2. Power plants

Fig. 1 shows the schematic layout of regular power plants used at LTHR and ITHR respectively for (a) ORC and (b) SRC. The ORC is selected with a regenerator OFC at LTHR and SRC is selected with a deaerator. Power generation from the flashing cycle and KCS are compared with the regular power plants at both heat recovery levels. Fig. 2 shows the details of the flash cycle used with steam or organic fluid. If steam is the working fluid with the flash, the cycle is called SFC, and with the organic fluid it is known as OFC. The hot streams of gases are supplied to the HRSG/HRVG. In a fuel fired power plant, the combustion gas temperature is high and so the pressure is high, at near to or above critical pressure. The super heated vapor at turbine inlet is expanded in the turbine by mixing saturated vapor from the flashers. The expanded vapor is completely condensed into saturated liquid. The vapor in turbine expands at four pressure stages viz. high pressure (HP), intermediate pressure (IP), low pressure (LP) and condenser pressure. The IP and LP pressures are also known as high pressure flashing (HPF) and low pressure flashing (LPF) respectively. The hot gas temperature decreases in HRSG/HRVG with a rise in liquid temperature and generation of superheated vapor. The pressurized hot water is collected from the HRSG/HRVG for flashing. The mass is proportional to the vapor capacity in the boilers. In the HP flasher, the liquid is flashed into wet vapor. The wet fluid consists of saturated liquid and saturated vapor. The saturated vapor is only supplied to the turbine. The liquid part of the wet fluid is connected to a second flasher or LPF. The liquid from the HP flasher is flashed again into wet vapor with a separation of liquid and saturated vapor. The expanded vapor is fully condensed into liquid using circulating water in a vapor condenser. The specialty of a flasher in a plant is the vapor addition to the turbine. Additional power can be generated with pressurized liquid flashing. It also enbales greater heat recovery due to the increased heat load in the economizer part of HRSG/HRVG. But heat recovery reduces cycle thermal efficiency, as efficiency is focused on minimum heat recovery with maximum power generation. The maximum efficiency condition is applicable for a power plant with a fuel firing system. The maximum power generation condition is suitable for waste heat recovery plants. Therefore, the power output in a flashed system is greater than the power generation from a regular plant. The additional power in the flash cycle is generated with a penalty in thermal efficiency. In ORC and OFC, R124 is used at LTHR and ammonia is selected as a working fluid at ITHR due to its suitability and greater power production.

Fig. 3a shows the working of KCS at LTHR. It is similar to a Rankine cycle with a modification made to achieve heat recovery with variable temperature and condense the working fluid without difficulty. The working fluid is a zeotropic mixture of ammonia and water. The temperature of the working fluid varies in the phase change, i.e., boiling and condensation. In HRVG, the liquid solution converts into a liquid-vapor mixture. In the separator, the vapor separates for the turbine. After expansion, the working fluid becomes a rich concentration and would not permit condensation at the atmospheric condition. Therefore it needs to be diluted with water or a weak solution from the separator before the condenser. Therefore, the total cycle facilitates effective heat recovery and easy condensation with the adoption of a separator. Shankar Ganesh and Srinivas [16] analyzed the KCS with low temperature solar thermal heat recovery. Fig. 3b shows the KCS at ITHR. The super heated vapor is expanded in the turbine and rejects heat to the regenerator for preheating the solution. The specialty of this KCS is operation without throttling devices. After the separator, the mixture is diluted by mixing the weak solution with the vapor for full condensation at the sink temperature. The pumped solution is preheated before the boiler by internal heat regenerators (HE_1 and HE_2) and external heat regenerators (HE_1 and HE_2). The external heat is distributed according to the fluid temperatures and arranged from HE_1 to HE_6 and finally the fluid is supplied to superheater. Shankar Ganesh and Srinivas [17] also addressed the operational conditions of KCS at intermediate temperature solar thermal heat recovery.

Thermal efficiency,

$$\eta_1 = \frac{W_{net}}{Q_{\sup ply}} \times 100 \tag{1}$$

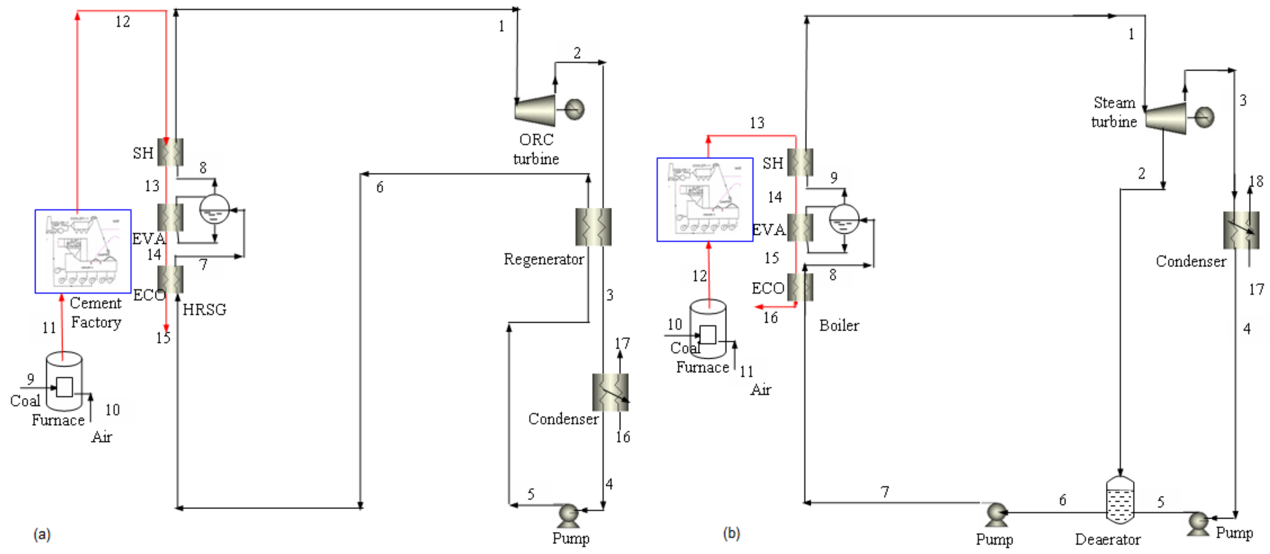

Figure 1: Regular power plants (a) ORC and (b) SRC

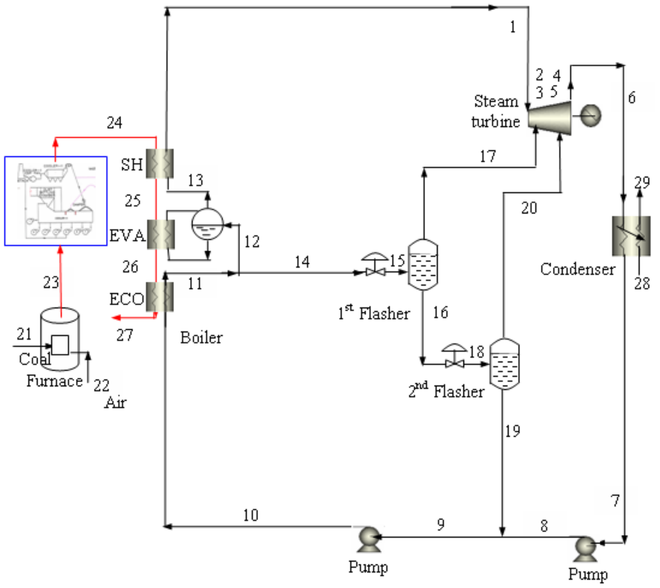

Figure 2: Flash cycle configured to OFC and SFC

Exergy efficiency,

$$\eta_2 = \frac{E_{hotgas} - \sum I_{total}}{E_{hotgas}} \times 100 \tag{2}$$

3. Results and Discussions

Fig. 4 shows the influence of HPF and LPF locations in OFC at LTHR. The LPF temperature is changed between condenser saturation temperature and HPF saturation temperature. Similarly, HPF temperature is varied between LPF saturation temperature and HRSG saturation temperature. The (a) power, (b) exhaust gas temperature, (c) cycle energy efficiency and (d) exergy efficiency of OFC are plotted with a change in LPF and HPF temperature ratios. The power, exhaust gas temperature, cycle energy efficiency and exergy efficiency increase with the increase in LPF and HPF. The total change in exhaust gas temperature is limited to 1°C with change in LPF and HPF. There is little change in exhaust gas temperature with temperature ratios. Heat recovery decreases with the increase in HPF and LPF temperature ratios, due to increased exhaust gas temperature. The increased HPF and LPF temperature increases the vapor pressure and temperature supplied to the turbine. It increases power with the rise in both LPF and HPF. Since the temperature ratio influences power more than the heat recovery, energy efficiency increases with the increase in both temperature ratios. Exergy efficiency is more sensitive than energy efficiency to the above stated changes.

Fig. 5 shows changes in the plant's power with HPF temperature ratio and LPF temperature ratio with (a) SFC and (b) OFC at ITHR. The cycle boiler and condenser temperature difference in the steam cycle is more than ORC. Therefore a wide range in steam temperature ratio is selected. A limited range is only selected in OFC. The optimum LPF temperature ratio is displaced left (condenser side) with the increase in HPF temperature ratio. The optimum HPF temperature ratio is increasing, reaching maximum and decreasing with an increasing HPF ratio. At 0.5 HPF temperature ratio, the plant output is at a maximum in SFC. The corresponding LPF temperature ratio is 0.5 at this condition. Therefore the recommended temperature ratio is 0.5 for both LPF and HPF. The power output is decreasing with the increase in HRSG pressure. At a fixed pinch point, the exhaust gas temperature increases with the increase in HRSG pressure. It reduces the heat recovery with the increase in HRSG pres-

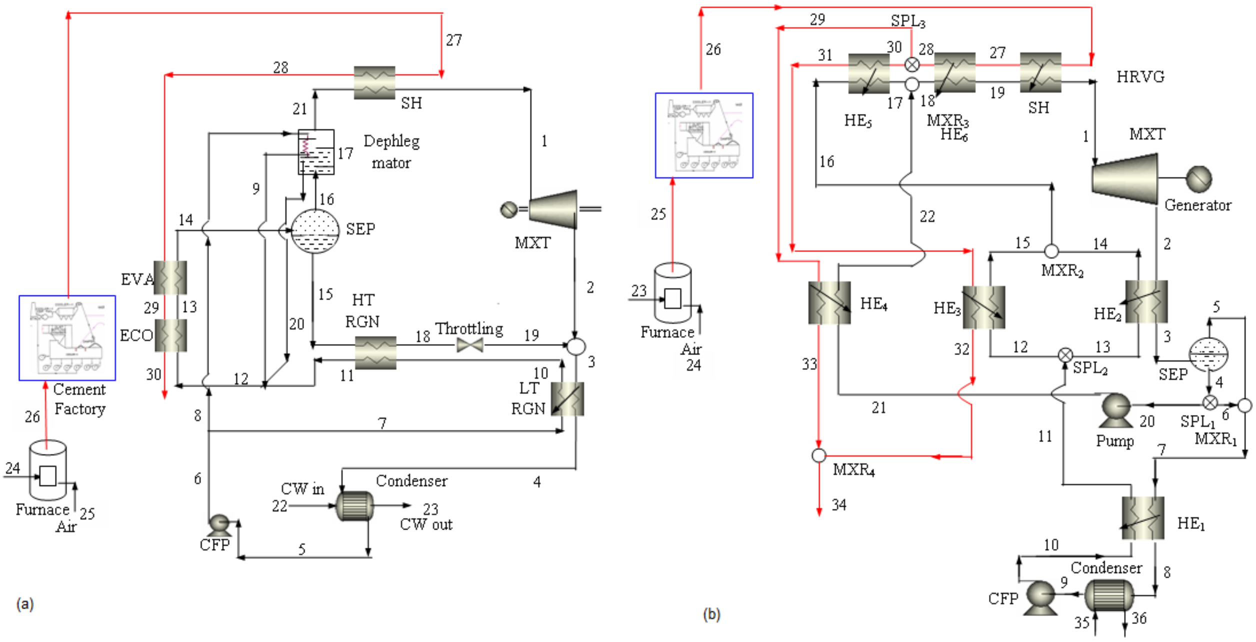

Figure 3: KCS with (a) LTHR and (b) ITHR

sure. Low HRSG pressure is preferable for a heat recovery based power plant. The HRSG pressure is fixed at 15.7 bar with reference to the operational data of the case study. In the case of ORC, the LPF can be selected above the ratio 0.5. The recommended LPF and HPF for ORC are 0.65 and 0.5 respectively.

The operational and performance conditions are plotted with the strong solution concentration by varying the hot gas supply temperature. The focused results are (a) power, (b) exhaust gas temperature, (c) cycle energy efficiency and (d) exergy efficiency. The power is reaching a maximum at every set of source temperature in increasing order of both temperature and concentration. The optimum strong solution concentration is increasing with the increase in gas temperature. For a typical hot gas inlet temperature of 125°C, the recommended strong solution concentration is 0.45. Similarly, the exergy efficiency trends follow approximately the power trends. The exhaust gas temperature decreases with the increase in solution concentration, but increases with the increase in gas temperature. The thermal efficiency of the plant is decreasing with the increase in the concentration and increasing with the increase in supply temperature.

Fig. 7 shows (a) cycle thermal efficiency-power and (b) plant exergy efficiency-exhaust gas temperature with a change in turbine inlet pressure and turbine inlet temperature at a fixed hot gas temperature. According to the permissible range, the turbine inlet temperature and pressure are varied as shown. To achieve the highest power generation, the plant demands low turbine inlet temperature with a penalty in thermal efficiency. The exergy efficiency is at a satisfactory level with flashing. At 48 bar of turbine inlet pressure, the power and exergy efficiency achieve a maximum. The exhaust gas temperature increases with the increase in turbine inlet temperature and pressure.

Table 1: ORC at LTRT

State	P, bar	T, °C	m, kg/s	h, kJ/kg	s, kJ/kg K
1	18	110	29.7	420	1.65
2	6.8	75.3	29.7	406	1.65
3	6.8	50	29.7	386	1.59
4	6.8	45	29.7	250	1.17
5	18	47.7	29.7	253	1.18
6	18	64	29.7	273	1.24
7	18	84.6	29.7	299	1.31
8	18	86.6	29.7	400	1.59
9	1.01	25	14.9	0	0
10	1.01	35	170	7.2	0.02
11	1.01	900	184	1006	1.54
12	1.01	125	184	104	0.3
13	1.01	107	184	84.7	0.25
14	1.01	107	184	84.7	0.25
15	1.01	103	184	80.5	0.24
16	1.01	25	121	0	0
17	1.01	33	121	33.4	0.11

Tables 1 to 8 show the material and energy balance results, furnished to give specific and detailed information to readers. Table 9 compares fluid flows, the heat duty of main heat exchangers and the performance of three power plants at LTHR and five power plants at ITHR. The fluid flow in the turbine and condenser and the source (HRSG/HRVG) and sink (condenser) are compared. The mass balance and energy balance results change with the change in power plant configuration, power enhancing techniques, working fluid and heat recovery level. For low temperature power plants, the working fluid at the inlet of the turbine is 29.65 kg/s in ORC and OFC plants. In the OFC plant the fluid at the exit of the OFC turbine is 37.48 kg/s as the extra flash vapor adds at

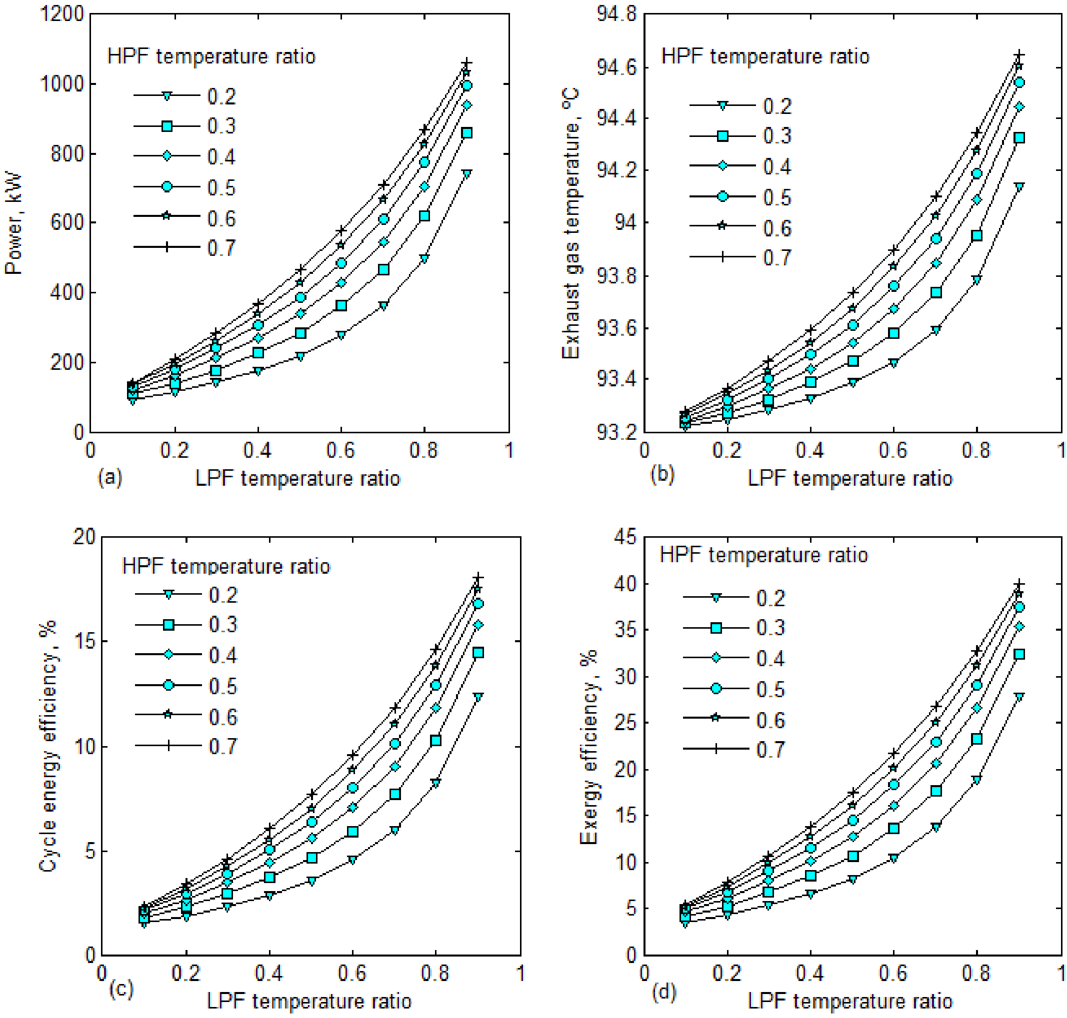

Figure 4: Effect of LPF and HPF temperature ratios in operational and performance conditions of OFC at LTHR

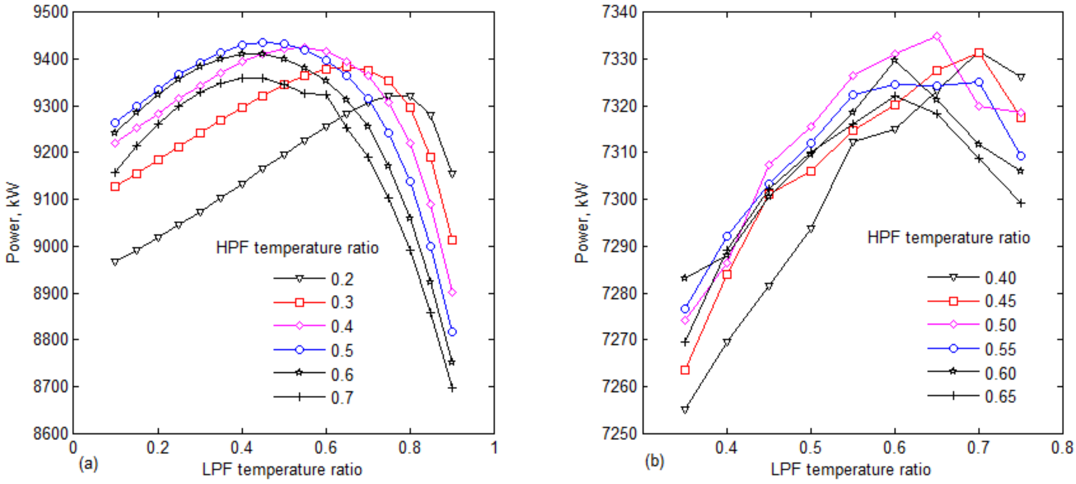

Figure 5: Effect of LPF and HPF temperature ratios on power generation with (a) SFC and OFC at ITHR

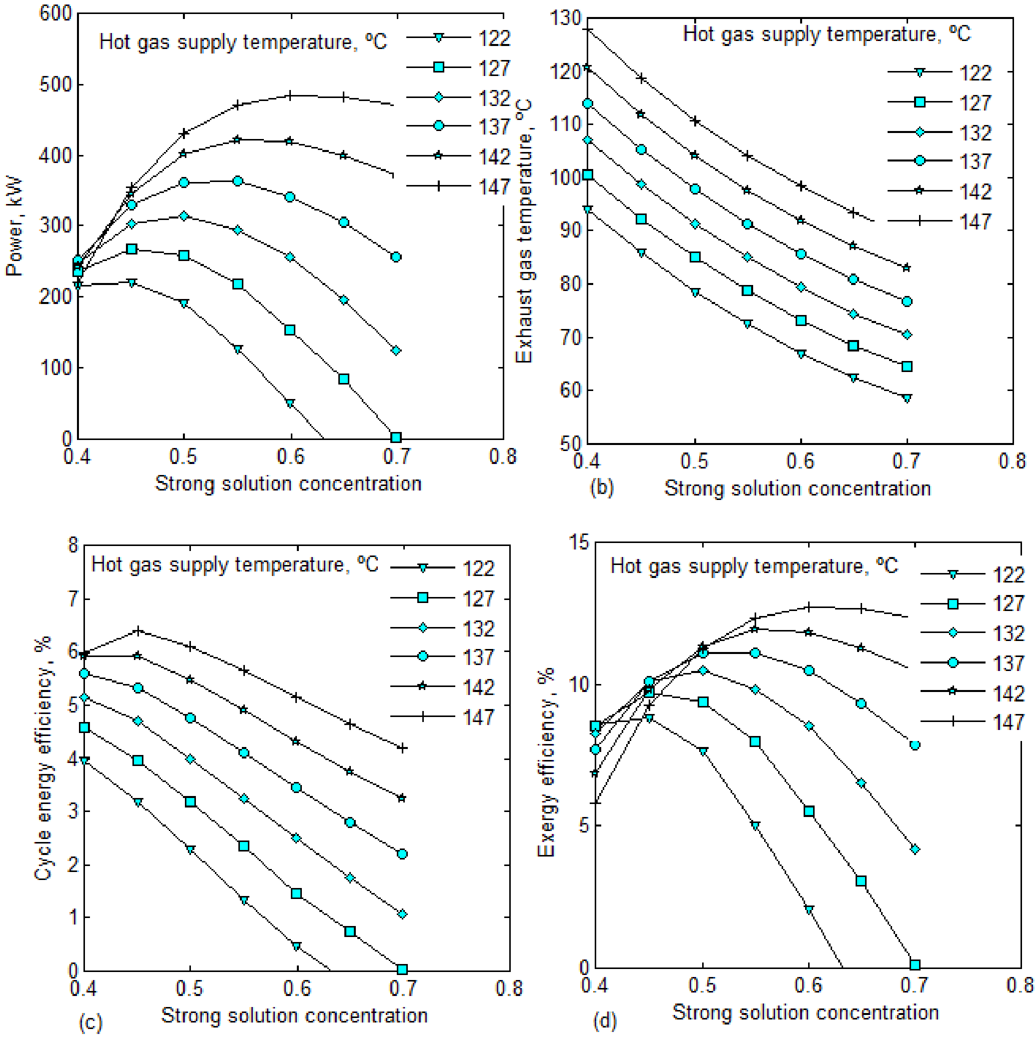

Figure 6: Effect of strong solution concentration with hot gas temperature with KCS at LTHR

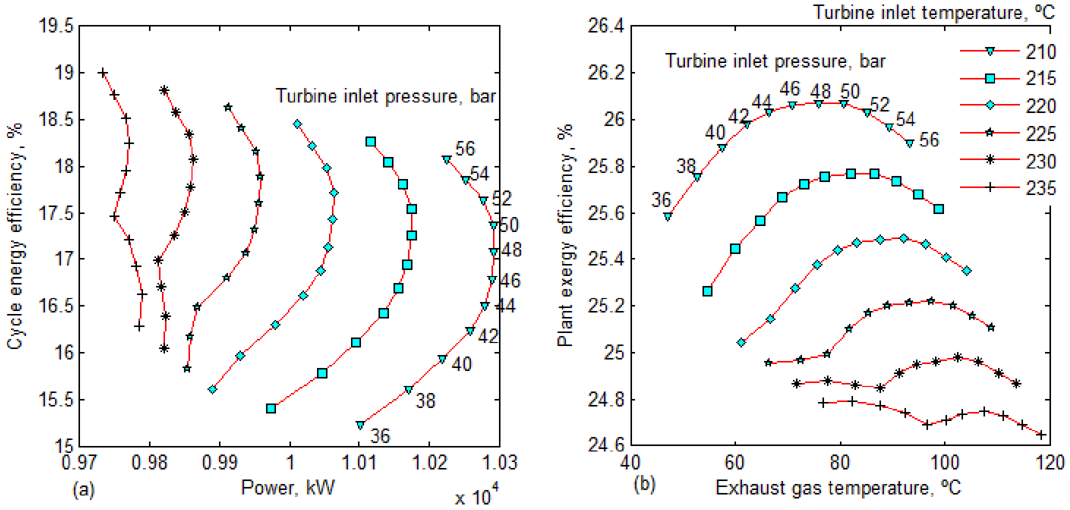

Figure 7: Effect of turbine inlet pressure and temperature performance of KCS at ITHR

Table 2: OFC at LTHR

State	P, bar	T,°C	m, kg/s	h, kJ/kg	s, kJ/kg K
1	18	110	29.7	420	1.65
2	13.4	97.8	29.7	416	1.65
3	13.4	94.7	33.9	413	1.64
4	9.69	83.1	33.9	408	1.64
5	9.69	80.9	37.5	406	1.64
6	6.8	69.5	37.5	401	1.64
7	6.8	45	37.5	250	1.17
8	9.69	47.7	37.5	253	1.18
9	9.69	48.7	59.3	258	1.18
10	18	51.4	59.3	257	1.19
11	18	84.6	59.3	299	1.31
12	18	84.6	29.7	299	1.31
13	18	86.6	29.7	400	1.59
14	18	84.6	29.7	299	1.31
15	13.4	73	29.7	299	1.31
16	13.4	73	25.4	284	1.27
17	13.4	73	4.22	395	1.59
18	9.69	59.3	25.4	284	1.27
19	9.69	59.3	21.8	266	1.22
20	9.69	59.3	3.61	389	1.59
21	1.01	25	14.9	0	0
22	1.01	35	170	7.2	0.02
23	1.01	900	184	1006	1.54
24	1.01	125	184	104	0.3
25	1.01	122	184	101	0.29
26	1.01	107	184	84.7	0.25
27	1.01	93.6	184	71.1	0.21
28	1.01	25	170	0	0
29	1.01	33	170	33.4	0.11

the intermediate states. In KCS out of 11.25 kg/s fluid, only 3.62 kg/s is available for turbine expansion. The fluid flow rate in the intermediate temperature power plant differs from the low temperature plants. The fluid flow in the steam turbine is very low compared to the organic turbine and Kalina turbine at the same heat source temperature and flow. More fluid is generated for the KCS turbine. Since flashing increases the condenser load, the circulating water flow also increased in OFCs at the two temperature levels. But the KCS at LTHR and LTHR demands the highest water circulation in the condenser as the working fluid temperature varies with phase change. The HRVG load with LTHR is greater with KCS and lower with ORC. The same is reflected in condenser duty. For intermediate temperature heat recovery, the heat supply to the steam power plant is increased from 33.77 MW to 41.77 MW with the change in configuration from regular to flashing. Similarly, the condenser duty also increased from 25.06 MW to 31.95 MW from SRC to SFC. The exhaust gas temperature is at a minimum and maximum respectively with KCS and ORC at low temperature source. The exhaust gas temperature is increased in OFC compared to ORC as the HRVG load increased, but not more than the KCS. In the low temperature power plants, the highest thermal efficiency (7.1%) is achieved with the ORC and the highest exergy efficiency (14.58%) is achieved with the OFC. Therefore KCS is not competitive with the organic group plant due to low efficiency and lower power. In steam power plants, due to increased heat addition with SFC, the exhaust gas temperature is decreased from 193°C to 152.5°C. The extra steam in the turbine increases the power from 8.34 MW to 9.43 MW. The increase in heat addition is more than the power rise, hence thermal efficiency decreases from 24.72% to 22.58%. But due to the effective use of waste heat, the exergy efficiency increases from 37.76% to 42.67%. Therefore SFC is suitable than the SRC to gain more power. These results repeat in ORC plants. The amount of heat supply and heat rejections are greater compared to steam power plants. But the power from ORC and OFC is less than from the SRC and SFC. The power increment in the steam power plant with flashing is more than the ORC. Due to large heat exchangers in ORC and OFC, the cost of the plant is higher than it is with the steam power plant. The exhaust gas temperature is too low in ORC plants compared to the others. The heat supply and heat rejection are intermediate between ORC and OFC. The power generation in KCS is more than with other plants because of the zeotropic mixture. The energy efficiency of KCS is not competitive with the steam power plants. The exergy efficiency of KCS is more than the other four plants due to close temperature profiles in the heat exchangers. Even though KCS is not competitive at low temperature thermal plants with organic cycles, it is well justified at the intermediate temperature plant over the other plants. Finally, OFC is at LTHR and KCS at ITHR are recommended with a view to achieving greater power and exergy efficiency.

4. Conclusions

Three power plants (ORC, OFC, KCS) at LTHR and five power plants (SRC, SFC, ORC, OFC, KCS) at ITHR were studied and compared in order to select the best one respectively at low and intermediate temperatures. Focus was given to maximize power production from waste heat recovery. KCS is the the competitive solution at the low temperature level, but it performswell at the intermediate temperature. OFC and KCS are recommended respectively at low and intermediate heat recovery temperatures as they achieve the highest power and exergy efficiency.

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Table 3: KCS at LTHR

State	P, bar	T, °C	x	m, kg/s	h, kJ/kg	s, kJ/kg K
1	11.1	110	0.95	3.62	1554	5.08
2	6	69.5	0.95	3.62	1470	5.13
3	6	82.1	0.5	10.9	637	2.51
4	6	80.7	0.5	10.9	614	2.45
5	6	45	0.5	10.9	-37.8	0.51
6	11.1	46.3	0.5	10.9	-31.7	0.53
7	11.1	46.3	0.5	9.97	-31.7	0.53
8	11.1	46.3	0.5	0.94	-31.7	0.53
9	11.1	93	0.5	0.94	544	2.17
10	11.1	51.6	0.5	9.97	-7.55	0.6
11	11.1	61.9	0.5	9.97	39.7	0.74
12	11.1	67.6	0.5	11.3	83	0.87
13	11.1	67.6	0.5	11.3	85.6	0.88
14	11.1	108	0.5	11.3	801	2.86
15	11.1	108	0.29	7.28	287	1.38
16	11.1	108	0.9	3.96	1601	5.18
17	11.1	92.2	0.9	3.96	1464	4.82
18	11.1	93.6	0.29	7.28	222	1.21
19	6	86.6	0.29	7.28	225	1.22
20	11.1	92.2	0.36	0.34	192	1.17
21	11.1	92.2	0.95	3.62	1509	4.96
22	1.01	25	0	213	0	0
23	1.01	33	0	213	33.4	0.11
24	1.01	25	0	14.9	0	0
25	1.01	35	0	170	8.18	0.03
26	1.01	900	0	184	1006	1.54
27	1.01	125	0	184	104	0.3
28	1.01	124	0	184	103	0.3
29	1.01	82.6	0	184	59.6	0.18
30	1.01	82.4	0	184	59.4	0.18

Table 4: RC at ITHR

State	P, bar	T, °C	m, kg/s	h, kJ/kg	s, kJ/kg K
1	15.7	345	12.3	3136	7.06
2	0.75	91.6	0.96	2642	7.4
3	0.1	45	11.3	2401	7.59
4	0.1	45	11.3	188	0.64
5	0.75	46	11.3	193	0.65
6	0.75	91.6	12.3	384	1.21
7	15.7	92.8	12.3	389	1.22
8	15.7	195	12.3	832	2.29
9	15.7	200	12.3	2791	6.42
10	1.01	25	14.9	0	0
11	1.01	308	170	7.2	0.02
12	1.01	900	184	0	1.54
13	1.01	360	184	360	0.8
14	1.01	339	184	337	0.77
15	1.01	220	184	206	0.53
16	1.01	193	184	177	0.47
17	1.01	25	749	0	0
18	1.01	33	749	33.4	0.11

Table 5: SFC at ITHR

State	P, bar	T,°C	m, kg/s	h, kJ/kg	s, kJ/kg K
1	15.7	345	12.3	3136	7.06
2	4.59	220	12.3	2899	7.19
3	4.59	213	13.5	2885	7.16
4	0.9	96.8	13.5	2645	7.32
5	0.9	96.8	14.6	2647	7.33
6	0.1	45	14.6	2383	7.54
7	0.1	45	14.6	188	0.64
8	0.9	46	14.6	193	0.65
9	0.9	66.9	24.6	280	0.92
10	15.7	67.9	24.6	284	0.93
11	15.7	195	24.6	832	2.29
12	15.7	195	12.3	832	2.29
13	15.7	200	12.3	2791	6.42
14	15.7	195	12.3	832	2.29
15	4.59	149	12.3	832	2.49
16	4.59	149	11.1	626	1.83
17	4.59	149	1.19	2744	6.85
18	0.9	96.8	11.1	626	1.87
19	0.9	96.8	10	406	1.27
20	0.9	96.8	1.08	2671	7.39
21	1.01	303	14.9	0	0
22	1.01	308	170	7.2	0.02
23	1.01	900	184	1006	1.54
24	1.01	360	184	360	0.8
25	1.01	339	184	337	0.77
26	1.01	220	184	206	0.53
27	1.01	152	184	133	0.37
28	1.01	25	956	0	0
29	1.01	33	956	33.4	0.11

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Table 6: ORC at ITHR

State	P, bar	T, °C	m, kg/s	h, kJ/kg	s, kJ/kg K
1	60.00	345.00	30.55	2078.20	5.31
2	17.93	239.81	30.55	1821.37	5.42
3	17.93	215.83	30.55	1758.90	5.30
4	17.93	45.00	30.55	216.03	0.74
5	60.00	47.88	30.55	230.62	0.78
6	60.00	59.98	30.55	293.09	0.97
7	60.00	95.45	30.55	487.64	1.50
8	60.00	97.45	30.55	1243.37	3.89
9	1.01	303.15	14.86	0.00	0.00
10	1.01	35.00	169.61	7.20	0.02
11	1.01	900.00	184.45	1006.40	1.54
12	1.01	360.00	184.45	359.60	0.80
13	1.01	234.57	184.45	221.32	0.56
14	1.01	117.45	184.45	96.15	0.28
15	1.01	86.75	184.45	63.93	0.19
16	1.01	25.00	1409.58	0.00	0.00
17	1.01	33.00	1409.58	33.44	0.11

Table 7: OFC at ITHR

State	P, bar	T, °C	m, kg/s	h, kJ/kg	s, kJ/kg K
1	60	345	30.6	2078	5.31
2	45.3	318	30.6	2009	5.32
3	45.3	299	32.9	1955	5.24
4	33.5	271	32.9	1887	5.25
5	33.5	257	35.2	1848	5.18
6	17.9	204	35.2	1727	5.23
7	17.9	45	35.2	216	0.74
8	33.5	46.7	35.2	225	0.76
9	33.5	56.8	61.1	277	0.92
10	60	59.2	61.1	287	0.93
11	60	95.5	61.1	488	1.5
12	60	95.5	30.6	488	1.5
13	60	97.5	30.6	1243	3.89
14	60	95.5	30.6	488	1.5
15	45.3	83.9	30.6	488	1.53
16	45.3	83.9	28.2	422	1.33
17	45.3	83.9	2.37	1268	3.94
18	33.5	70.3	28.2	422	1.35
19	33.5	70.3	25.9	347	1.12
20	33.5	70.3	2.24	1284	4
21	1.01	303	14.9	0	0
22	1.01	35	170	7.2	0.02
23	1.01	900	184	1006	1.54
24	1.01	360	184	360	0.8
25	1.01	235	184	221	0.56
26	1.01	117	184	96.2	0.28
27	1.01	53.9	184	29.8	0.1
28	1.01	25	1589	0	0
29	1.01	33	1589	33.4	0.11

Table 8: KCS at ITHR

State	P, bar	T, °C	x	m, kg/s	h, kJ/kg	s, kJ/kg K
1	50	225	0.8	58.2	1920	5.2
2	12.1	123	0.8	58.2	1716	5.36
3	12.1	75	0.8	58.2	1000	3.48
4	12.1	75	0.47	20.7	99.4	0.92
5	12.1	75	0.98	37.5	1429	4.71
6	12.1	75	0.47	7.12	99.4	0.92
7	12.1	75	0.9	44.6	1262	4.23
8	12.1	68.9	0.9	44.6	1202	4.06
9	12.1	35	0.9	44.6	92.6	0.55
10	50	37.5	0.9	44.6	106	0.57
11	50	49.3	0.9	44.6	163	0.75
12	50	49.3	0.9	3.26	163	0.75
13	50	49.3	0.9	41.3	163	0.75
14	50	116	0.9	41.3	1172	3.58
15	50	116	0.9	3.26	1172	3.58
16	50	116	0.9	44.6	1172	3.58
17	50	149	0.9	44.6	1579	4.51
18	50	149	0.8	58.2	1348	3.96
19	50	182	0.8	58.2	1787	4.93
20	12.1	75	0.47	13.6	99.4	0.92
21	50	77.2	0.47	13.6	112	0.94
22	50	149	0.47	13.6	557	2.08
23	1.01	303	0	19.3	0	0
24	1.01	35	0	167	8.32	0.03
25	1.01	900	0	186	1021	1.56
26	1.01	360	0	186	363	0.81
27	1.01	323	0	186	322	0.74
28	1.01	199	0	186	185	0.49
29	1.01	199	0	52.9	650	1.71
30	1.01	199	0	133	258	0.68
31	1.01	71.5	0	133	67.5	0.21
32	1.01	47.9	0	133	33	0.11
33	1.01	92.2	0	52.9	247	0.75
34	1.01	60.5	0	186	36.8	0.12
35	1.01	25	0	1480	0	0
36	1.01	33	0	1480	33.4	0.11

Table 9: Summary of SRC, SFC, ORC, OFC and KCS power plant results at low temperature heat source of 125℃ and intermediate temperature heat source 360℃ with 500000 Nm^3/h gas flow

Heat recovery	LT heat recovery			IT heat recovery				
Power plant	ORC	OFC	KCS	SRC	SFC	ORC	OFC	KCS
Power plant	ORC	OFC	KCS	SRC	SFC	ORC	OFC	KCS
Turbine inlet flow rate, kg/s	29.7	29.7	3.62	12.3	12.3	30.6	30.6	58.2
Condenser circulating water, kg/s	121	170	213	749	956	1410	1589	1480
HRSG/HRVG capacity, kW	4364	6088	8076	33770	41780	54540	61250	59470
Condenser capacity, kW	4035	5680	7115	25060	31950	47140	53550	48410
Net power, kW	310	387	231	8348	9433	7081	7331	10180
Exhaust gas temperature, °C	103	93.6	82.4	193	153	86.8	51.8	61.1
Thermal efficiency, %	7.1	6.35	2.86	24.7	22.6	13	12	17.1
Exergy efficiency, %	11.7	14.6	8.7	37.8	42.7	32	33.2	46.1