SELECTION OF LOW TEMPERATURE THERMAL POWER CYCLES

by

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In today's world, we are facing the problem of fossil fuel depletion along with its cost continuously increasing. Also, it is getting difficult to live in a pollution free environment. Solar energy is one of the most abundantly and freely available form of energy. Out of the various ways to harness solar energy, solar thermal energy is the most efficient as compared to photovoltaic technology. There are various cycles to convert the solar thermal energy to mechanical work, but Kalina cycle is one of the best candidates for high efficiency considerations. Therefore, the authors have proposed a novel Kalina cycle having the double separator arrangements to increase the amount of ammonia vapors at the inlet of turbine, and hence have tried to minimize the pumping power for double separator Kalina cycle by reducing the fraction of gas/vapors through it. Here, in this paper we have tried to compare ORC, Brayton cycle, and double separator Kalina cycle for low temperature heat extraction from parabolic trough collectors having arc-circular plug with slits. The effect of different operating conditions, like the number of parabolic trough collectors, mass-flow rate of fluids in different cycles, pressure difference in turbine are analyzed. The effect of these different operating conditions on different parameters like net work done, heat lost by condenser, thermal efficiency and installation cost per unit kW for double separator Kalina cycle, ORC, and Brayton cycle are studied.

Key words*: performance comparison of different cycles, Brayton cycle, double separator Kalina cycle, ORC*

Introduction

There are various means of extracting useful mechanical work with the help of different proposed cycles. Many of them have investigated about Rankine cycle (RC), ORC, Brayton cycle (BC), and Kalina cycle (KC) for different kinds of waste heat extraction and also even from solar thermal energy. This paper deals with the extraction of heat by parabolic trough collectors (PTC), and it is the most vital part of a solar thermal power plant. Therefore, the designing and analysis of PTC are critically important for all kinds of cycles to be used in thermal power plants. Li *et al.* [1] proposed that R123 is more suitable than R245fa for low operating pressure and high efficiency. Eldean *et al.* [2] tried to compare the performance of Stirling and Brayton cycles for different working fluids. It can be seen that for low temperature and low pressure ratios, the monoatomic gases showed better results as compared to other fluids. Bahrampoury

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and Behbahaninia [3] showed that as compared to single separator KC, the dual separator KC system's exergy efficiency improved by 18% with an improvement of 17% in optimum mass fraction of ammonia. Therefore, from aforementioned literature; the type of PTC, modified KC *i.e.* Double separator Kalina cycle (DS-KC) and the fluids for BC and ORC were decided. Here, krypton had been considered for BC; whereas for ORC, R123 had been taken into account. Many researchers had compared the performance of different kinds of cycles, out of which only KC, BC, and ORC had been considered under this study. Bombarda *et al.* [4] showed that for medium and low temperature levels, the performance of ORC was found to be better than KC. The power generated from the exhaust heat recovery of Diesel engine with a mass-flow rate of 35 kg/s and at 346 °C for KC and ORC were 1615 kW and 1603 kW, respectively. The power generated by these two cycles are almost equal in value; and to achieve it, the maximum pressure requirement of Kalina was 100 bar as compared to ORC with 10 bar. Rodriguez *et al.* [5] studied that the KC was 18% more efficient than that of ORC, with NH_3-water mixture (84% NH_3 and 16% water mass fraction) and R290 as fluids used in these cycles, respectively. Also, the levelized cost for KC was found to be 1.2 times lesser than that of ORC. Shokati *et al.* [6] proposed that the maximum value of electrical power produced by duel pressure ORC is about 43.48% more than that of KC, while that of the minimum value for per unit cost of energy produced with KC was about 52.09% lesser as compared to duel pressure ORC. Wang *et al.* [7] discussed that for any kind of waste heat, the heat recovery efficiency of KC was better than as compared to ORC; whereas the thermal efficiency of KC had been found to be less than that of ORC. Also, it had been found that if the ratio of heat above the moist point and below it was more than 0.2, then KC showed better performance than that of ORC; while for convex heat ORC performed better. Fiaschi *et al.* [8] studied that for medium temperature sources; the ORC with R123 performed better than KC, with levelized cost less than 3% than that of KC. But for low temperature sources, KC performed better than that of ORC, with levelized cost 24-43% lesser as compared to ORC operating with different working fluids. Meng *et al.* [9] investigated that the net power output of transcritical CO_2 cycle was more than that of KC and ORC, but the economic performance of it was between KC and ORC. Now, from literature review; low critical temperature fluids have been selected by the for BC and ORC. Less number of literatures are available regarding the performance comparison of KC and ORC, while a nice number of literatures are available regarding the performance estimation of combined cycle by considering one of it as topping or bottoming cycles. Almost no literatures are available regarding the performance and economic comparison of KC, BC and ORC by considering heat extraction with any type of solar collectors. Based upon the literature gap, KC with double separator has been proposed to minimize the pumping power required by reducing the amount of vapour content passing through it. In this paper, authors have compared solar assisted DS-KC, BC and ORC by using PTC as thermal energy transferring system.

System's description

A PTC of arc-circular plug with slits has been used for the harnessing of solar thermal energy. The energy and installation equations that are encrypted within the Chemcad/Dwsim are depicted in tabs. 1 and 2, respectively. Ansys-AIM 2019R1 software has been used for the validation and simulation of PTC as depicted in tab. 3. Therminol-vp1 is used as the heat transfer fluid in PTC. The isentropic efficiency of the turbine and pumps are assumed to be 75%, and a solar flux of 933.7 W/m^2 has also been considered constant for all cases. This paper presents a double separator KC and its performance comparison with ORC and BC. The cycles are designed and validated in open source software, also some arrangements of it has been tested in Chemcad.

Table 1. Energy equations used in the solar thermal plant's system analysis are below [10-12]

The useful thermal energy extraceby PTC was evaluated $Q_{pr} = mC_p(T_{out} - T_{in})$	Solar energy available was evaluated $Q_{ab} = Ar_{ap}I_{bm}$
Power developed by the turbine was evaluated $m_f(h_{tu,in} - h_{tu,out})\eta_{tur}$	Power consumption by compressor was evaluated $P_{co} = m_f(h_{co,in} - h_{co,out})\eta_{comp}$
The power consumption by pumps was evaluated $P_{pu} = m_f(p_{pu,in} - p_{pu,out})/\rho_f\eta_{pu}$	The heat exchanger is simulated based on simple weighted model $\dot{q}_{HE} = m_{w,f}(H_{HE,i+1} - H_{HE,i})$

Table 2. Installation cost functions and equations for different subsystems are below [11, 12]

Turbine: $C_{tu} = (576.1/397) \times 3.5 \times C_{tu}^0$ $\log C_{tu}^0 = 2.7051 + 1.4398\log\dot{W}_{tu} - 0.1776(\log\dot{W}_{tu})^2$	Heat-exchangers: $C_{HE} = (576.1/397)(1.63 + 1.66 \times 1)C_{HE}^0$ $\log C_{HE}^0 = 4.3247 - 0.3030\log A_{HE} + 0.1634(\log A_{HE})$
Separator: $Z_{se} = 5.93 \times C_{se}^0$ $\ln C_{se}^0 = 3.49 + 0.448\ln V_{se} + 0.10789(\ln V_{se})^2$ $V_{se} = 81.24D_i^3$	Pump: $C_{pu} = (576.1/397)(1.89 + 1.35 \cdot 1.5)C_P^0$ $\log C_{pu}^0 = 3.3892 + 0.0536\log\dot{W}_{pu} + 0.1538(\log\dot{W}_{pu})^2$

Table 3. Validation of numerical simulation with Dudley's work [13]

Case	DNI	Mass-flow rate [kgs^{-1}]	Inlet temperature [K]	Outlet temperature [K] LS-2 PTC	Outlet temperature [K] Numerical	Efficiency, η [%] Experimental	Efficiency, η [%] Numerical	ΔT [%]
1	933.7	0.6782	375.5	397.5	401.9675	72.071	72.024	-1.12
2	928.3	0.7205	471	493	491.0957	69.413	69.446	0.38
3	909.5	0.81	524.2	542.9	538.926	67.329	67.415	0.73

Characterization for double flash Kalina cycle

Figure 1 shows the line diagram of KC. The fluid (1) from outlet of pump [4] passes through the evaporator [5], which is exchanging heat with PTC; the outlet stream (2) from evaporator passes through a mixer [8] by mixing with a stream (4) which is an outlet of separator [3]. The outlet stream (12) of mixer [8] is inlet to separator [2], the vapour dominated outlet (3) from separator [2] is sent to inlet of turbine [1]. While other two outlets (6), (7) via expansion valves [9], [10], respectively from separator [2] and outlet (5) from condenser [6] are sent to separator [3], while the outlet (5) of turbine [6] is sent to the condenser [6]. The two outlets (9) , (10) from separator [3] are sent to mixer [7], the outlet (11) from mixer [7] is sent to pump [4]. In this paper two separator are used in KC with an arrangement of networks of stream, so that less power will be required by the pump. This can be achieved by letting less vapour stream to pass through the pump.

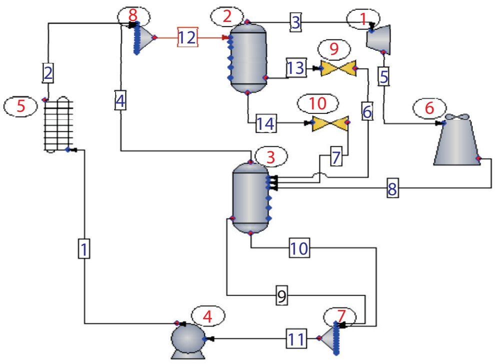

Figure 1. Line diagram of DS-KC

Characterization for Brayton and ORC cycle

Figure 2 shows the line diagram of BC, where krypton has been used as the heat transfer fluid. The fluid stream (3) from the outlet of condenser [3] passes through the compressor

[1]. The outlet stream (1) from the compressor [1] passes through the evaporator [4], the outlet stream (4) from evaporator [4] is sent to inlet to turbine [2]. Then, the outlet stream (2) from the turbine is sent to condenser [3]. Figure 3 shows the line diagram of ORC, where R123 has been used as the heat transfer fluid. The outlet fluid stream (1) from condenser [3] passes through a pump [1]. The outlet fluid stream (2) from the pump [1] is sent to evaporator [4], the outlet stream (3) from the evaporator [4] is sent to the inlet to turbine [2], then the outlet stream (4) from the turbine [2] is sent to inlet to condenser [3].

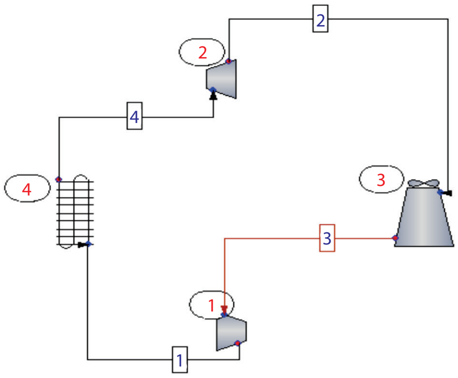

Figure 2. Line diagram of BC

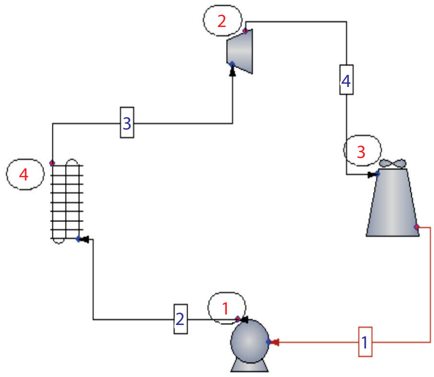

Figure 3. Line diagram of ORC

Calculation method

The thermodynamic model was developed by using two software, 1st is Dwsim (an open source software) and 2nd by using Chemcad. Both of these software have libraries from where you can drag and drop different components of your choice. Also, a good number of libraries are also there for different fluids having all properties associated with it. After selecting the fluids and desired models to be used in the simulation, all the components can be joined by using stream connectors and then simulation run is applied. Since, the fluids of our choice are hydrocarbons and therefore, all of their thermodynamic properties are predicted by using SRK methods depending upon the type of solving requirements as suggested by [14]:

$$P = \frac{\mathrm{R}T}{(V-b)} - \frac{b(T)}{V(V+b)} \tag{1}$$

The a and b parameters are given:

$$a(T) = \left[1 + \left(0.48 + 1.574\omega - 0.176\omega^2\right)\left(1 - T_\mathrm{r}^{0.5}\right)\right]^2, \quad b = 0.0864\left(\frac{\mathrm{R}T_c}{P_c}\right)$$

The thermal efficiency of the cycle is calculated:

$$\eta = \frac{P_\mathrm{tu} - P_\mathrm{pu/co}}{Q_\mathrm{pr}} \tag{2}$$

where R is the ideal gas universal constant, V – the for molar volume, T_c [k] – the for critical temperature, P_c [Pa] – the for critical pressure, ω – the for acentric factor for species, a – the parameter related to intermolecular forces, b – the parameter related to hard-sphere volume, P_tu [kW] – the power developed by turbine, $P_\mathrm{pu/co}$ [kW] – the power consumed by pump/compressor, and Q_pr [kW] – the net heat supplied by PTC.

Results and discussion

Thermodynamic model validation of DS-KC

The thermodynamic model of KC has been validated with the previous work on KC by Ogriseck [15], in this work author had maximized the generated electricity with recovery of heat without any demand of extra fuel for the existing plant. The calculations showed that the net efficiency of the KC plant is between 12.3-17.1 % depending on the cooling water temperature and mass fraction of ammonia in solution. Table 4 compares the results of present work with the previous work by Ogriseck [15]. Figure 4 shows the line diagram of KC, which was validated with KC by Ogriseck.

Table 4. Validation of the results of present work; (a) with the literature and (b) for KC

Stream	Temperature [°C]		Pressure [Pa]		Ammonia concentration [kg NH_3 per kg solution]		Mass-flow rate [kgs^{-1}]	
	(a)	(b)	(a)	(b)	(a)	(b)	(a)	(b)
1	8.15	8.0	4.6	4.6	0.82	0.82	16.8	16.8
2	8.15	8.0	35.3	35.3	0.82	0.82	16.8	16.8
3	44.0	41.0	34.3	34.3	0.82	0.82	16.8	16.8
4	60.65	63.0	33.3	33.3	0.82	0.82	16.8	16.8
5	111.842	116.0	33.3	32.3	0.82	0.82	16.8	16.8
6	116.15	116.0	32.3	32.3	0.97	0.97	12.83	11.4
7	43.05	43.0	6.6	6.6	0.97	0.97	12.83	11.4
8	44.123	46.0	6.6	6.6	0.82	0.82	16.8	16.8
9	30.19	30.0	5.6	5.6	0.82	0.82	16.8	16.8
10	116.5	116.0	32.3	32.3	0.63	0.50	3.97	5.4
11	44.123	46.0	31.3	31.3	0.36	0.50	3.97	5.4

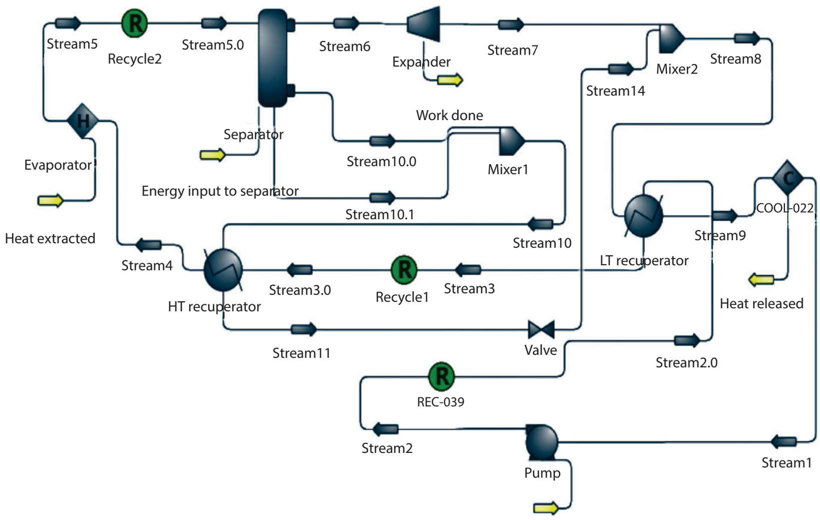

Figure 4. Validation with KC by Ogriseck [15]

Effect of number of PTC in series/heat addition

The effect of increase in number PTC has been studied by keeping the mass-flow rate as 1 kg/s, and inlet pressure to the turbine as 12 bar. It can be seen from fig. 5 that with increase in number of PTC-PS from 1-17 in numbers, the net power developed (NPD) by all the three cycles tends to increase. Because, increase in number of PTC leads to increase in the heat addition the cycles, and therefore, the enthalpy of the outlet stream from the evaporator also increases. Now, the increased enthalpy of the inlet stream to the turbine leads to increase the enthalpy of the outlet stream and power developed by the turbine. But the power developed by DS-KC is the highest than that of ORC and BC. One of the reasons is due to high critical temperature of NH_3-water mixture as compared to krypton and R245fa. The sequence of critical temperature of fluids in the order of decrement are NH_3-water mixture > krypton > R245fa. Therefore, the power developed by the turbine for these three cycles also follows the same sequence of decrement order as DS-KC > BC > ORC. Secondly, it is due to low power consumption required by KC pump; whereas the power consumption by compressor and pump of BC and ORC, respectively are high enough to decrease the NPD by cycles.

Figure 6 shows that with increase in number of PTC, the heat lost by the condenser (HLC) also increases. As discussed, increase in number of PTC the enthalpy of outlet stream also increases. Now, the increased enthalpy of the hot inlet stream to the condenser leads to increase in the enthalpy of cold outlet stream from the condenser. Therefore, with increase in the enthalpy of cold outlet stream from the condenser, heat loss from the condenser also increases.

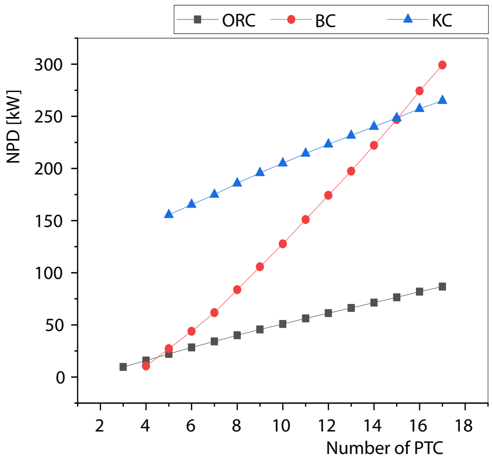

Figure 5. Variation of NPD with PTC

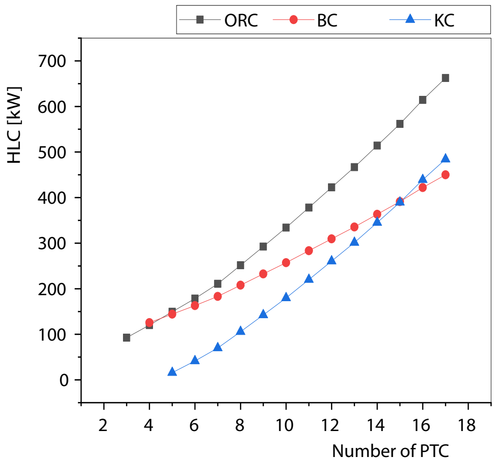

Figure 6 .Variation of HLC with PTC

Figure 7 shows that with increase in number of PTC, the thermal efficiency of DS-KC decreases while that of BC increases. Because, as it is evident from figs. 5 and 6 that with increase in number of PTC; the rate of NPD by BC increases as compared to rate of heat loss by its condenser. Whereas, in case of DF-KC; the rate of NPD by it decreases as compared to the rate of heat loss by its condenser with increase in number of PTC. In case of ORC the efficiency first increases from 8.074-13.93% with increase in number of PTC from 2-6; because up to 6 number of PTC, the rate of heat loss remains almost constant. But with increase in number of PTC beyond 6, the efficiency of ORC decreases to 11.578%; because beyond 6 numbers of PTC, the rate of heat loss increases as compared to rate of NPD by the ORC.

Figure 8 indicates that with increase in number of PTC in series, the installation cost per unit kW (ICPUK) for ORC, BC, and DS-KC decreases. This happens because; with increase in number of PTC in series, more amount of heat is supplied to the cycle. Due to increase in heat addition, the power developed by the turbine increases, and therefore, the NPD is enhanced with respect to the constant installation cost. But ICPUK of BC is the highest than that of ORC and DS-KC. The high installation cost of BC is only due to the installation cost of the compressor, as high power consumption is needed in increasing the pressure of a gas to a particular limit. Also, the rate of decrease of ICPUK for BC is the highest as compared to other two cycles. Because, as shown in fig. 5 the rate of increase of NPD by BC is greater as compared to DS-KC and ORC.

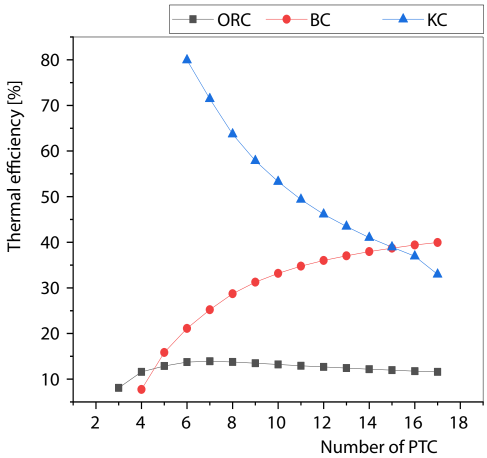

Figure 7. Variation of thermal efficiency *vs.* PTC

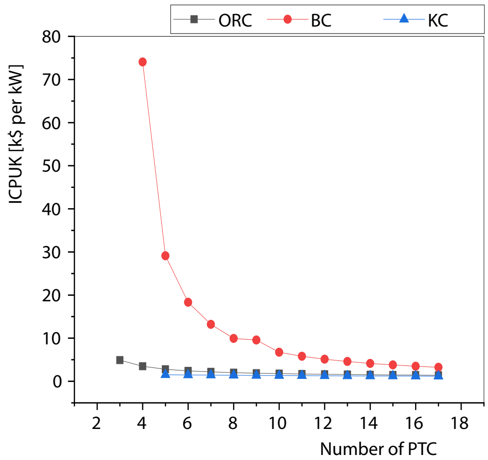

Figure 8. Variation of ICPUK *vs.* PTC

Effect of increase in mass-flow rate

The effect of increase in mass-flow rate (MFR) has been studied by keeping the number of PTC 17, that is heat input as 749.2487 kW and inlet pressure to the turbine of 12 bar. Figure 9 shows that with increase in MFR, the NPD by BC decreases while that for DS-KC increases. Whereas the NPD by ORC increases up to MFR of 3 kg/s. Because with increase in MFR, the concentration of vapors in the evaporator increases. This causes to increase in more heat addition from the evaporator to the fluid vapors. Therefore, it leads to increase in the enthalpy of inlet stream to the turbine, hence the power developed as well as the enthalpy of the outlet stream from the turbine also increases. But the NPD by BC decreases with increase in MFR, because work consumption by compressor increases with increase in MFR. Similarly, with an increase in MFR beyond 3 kg/s, the NPD by ORC decreases. This happens because with increase in MFR beyond 3 kg/s, the power consumed by ORC pump increases and this suppresses the effect of increase in power developed by turbine.

Also, it can be seen from fig. 10 that with increase in MFR the heat lost from condenser of DS-KC decreases, while that of BC increases. Whereas the heat lost from the condenser of ORC first decreases up to a MFR of 3.5 kg/s; after that with increase in MFR, it increases. Because, the power consumption by the pump of DS-KC is negligible as compared to power consumption by the compressor of BC. Also, the power consumption by the pump of ORC increases after a MFR of 3.5 kg/s. Therefore, with increase in MFR; the power consumption

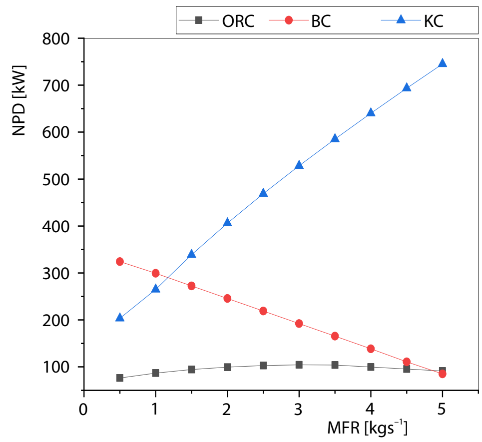

Figure 9. Variation of NPD *vs.* MFR

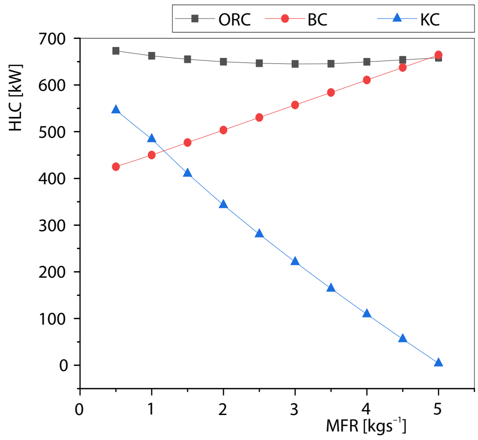

Figure 10. Variation of HLC *vs.* MFR

by the compressor and ORC pump (after 3.5 kg/s) increases, which in turn raises the enthalpy of the fluid system. Now, this increased enthalpy of the system carries more internal energy as compared to flow work; and a negligible amount of this internal energy is converted in the power by turbine. Power developed by turbine is mainly due to heat addition from the evaporator, other factors may contribute but have less impact as compared to former. Thus, the enthalpy of outlet stream from the turbine increases with increase in internal energy input from the compressor and pump (after 3.5 kg/s). Therefore, increase in enthalpy of the fluid due to increase in internal energy by the compressor and pump (after 3.5 kg/s) leads to increase the heat lost by condenser.

Figure 11 shows that with increase in MFR the efficiency of BC decreases while that of DS-KC increases. Whereas the efficiency of ORC increases with increase in MFR up to 3.5 kg/s; but after a MFR of 3.5kg/s, the efficiency of ORC decreases. As it is evident from figs. 9 and 10 that with increase in MFR; the heat loss by condenser of BC increases, whereas the NPD by it decreases and this is the cause of decrease in efficiency of BC. In case for DS-KC, with increase in MFR; the NPD by it increases, while the heat loss by its condenser decreases

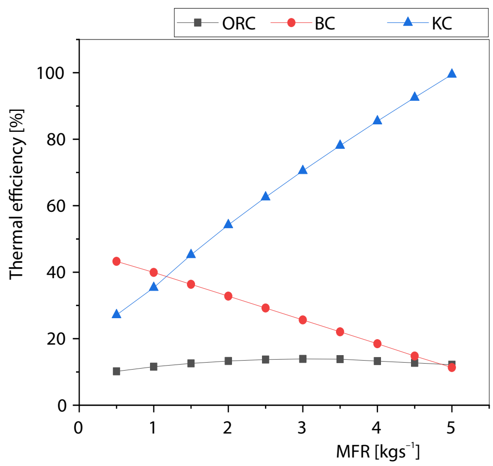

Figure 11. Variation of themal efficiency *vs.* MFR

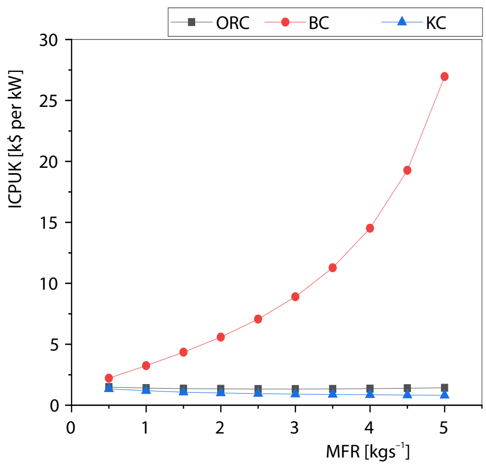

Figure 12. Variation of ICPUK vs MFR

and this reason is responsible for the increase in its efficiency. Whereas for ORC the rate of heat loss by condenser varies from negative to zero with increase in MFR from 0.5-3.5 kg/s. Beyond a MFR of 3.5 kg/s, the rate of heat loss from condenser of ORC increases from zero to positive. Therefore, the efficiency of ORC first increases with increase in MFR from 0.5-3.5 kg/s; but beyond 3.5 kg/s it decreases. But, with increase in MFR; the NPD by DS-KC increases more as compared to ORC. Also, the rate of heat loss by the condenser of DS-KC decreases more as compared to ORC with increase in MFR. Therefore, the efficiency of DS-KC is higher than ORC.

Figure 12 shows that with increase in MFR the ICPUK for BC increases. Because increase in MFR raises the quantity of gas to be compressed, which in turn requires a large compressor unit. Therefore, with increase in the requirement of large compressor unit directly affects the increment in cost. While that for DS-KC the ICPUK decreases with increase in MFR. Because with increase in MFR, the NPD by DS-KC increases by keeping a constant installation cost. Whereas the ICPUK for ORC first decreases up to a MFR of 3.5 kg/s, then beyond 3.5 kg/s it increases. Because the NPD by ORC increases up to a MFR of 3.5 kg/s by keeping a constant installation cost. Then beyond a MFR of 3.5 kg/s the NPD by ORC decreases for the same constant installation cost.

Effect of increase in pressure ratio

The effect of increase in pressure to the inlet of turbine has been studied, by considering a MFR of 1 kg/s and number of PTC as 17. Figure 13 shows that with increase in pressure, the NPD by ORC increases. While, the NPD by DS-KC and BC increases with increase in pressure up to 72 bar and 52 bar, respectively. This happens because with increase in shaft work to the compressor and pump, the flow work and internal energy of the inlet stream to the evaporator increases. Then, after getting heat from the evaporator, the enthalpy of the outlet stream from the evaporator increases again. Therefore, increase in enthalpy of the inlet stream to the turbine leads to increase the shaft work done by the turbine. But increase in pressure also leads to increase in shaft work needed by the pumps and compressors. As the power consumed by pumps with increase in pressure for DS-KC (before 72 bar) and ORC are not large enough to suppress the power developed by the turbine of these cycles; therefore, the NPD by the cycles increases. It can also be seen from the fig. 13, that the NPD by the DS-KC and BC decreases with increase in pressure beyond 72 bar and 52 bar, respectively. Because after increase in pressure beyond 72 bar for DS-KC and 52 bar for BC, the power needed by pump and compressor supersedes the rate of power developed by the turbine. Therefore, after a certain pressure value; there is a decrease in the NPD by DS-KC and BC.

Figure 14 shows that with increase in pressure the heat lost by condenser for ORC decreases. Whereas, with increase in pressure ratio; the heat lost by condenser for DS-KC decreases up to 72 bar and for BC decreases up to 52 bar, respectively. Because, increase in pressure ratio leads to increase in shaft work needed by compressor and pump; which in turn raises the quantity of only flow work in the enthalpy content of the outlet stream. The internal energy content of the outlet stream from the compressor and pump is only raised by the heat addition from the evaporator. Therefore, this increased internal energy and flow work leads to increase the enthalpy of the inlet stream to the turbine. But for DS-KC and BC the HLC increases after 72 bar and 52 bar, respectively. Because, after a certain value of pressure; the internal energy of the outlet stream from the pump (after 72 bar) and compressor (after 52 bar) also increases. Due to increase in internal energy, the temperature of the gas through the exit of the compressor and pump also increases, which in turn raises the internal energy of the inlet stream to the turbine.

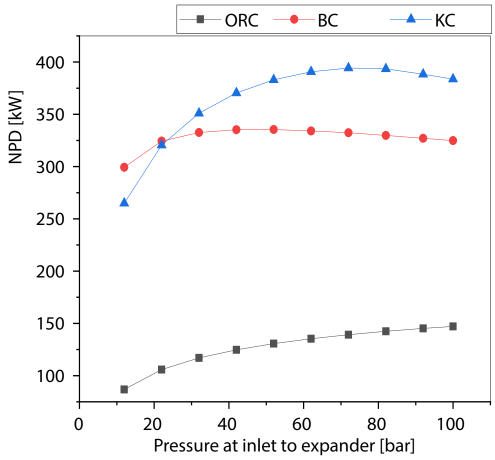

Figure 13. Variation of NPD *vs.* PIT

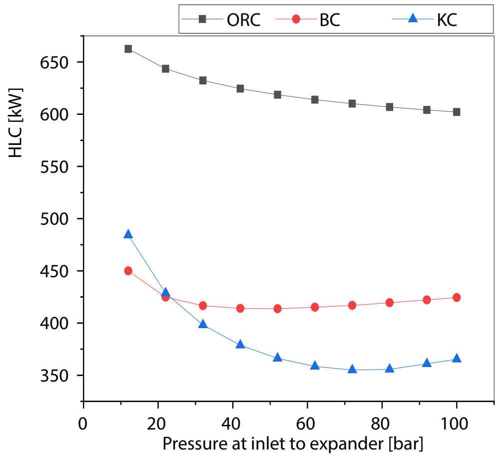

Figure 14. Variation of HLC *vs.* PIT

Therefore, after a certain pressure point this extra internal energy by the pump and compressor is lost in terms of HLC of DS-KC and BC.

Figure 15 shows that with increase in pressure the efficiency of ORC increases, while that of DS-KC and BC it increases up to 72 bar and 52 bar, respectively. As it is evident from Figusre 13 and 14 that with increase in pressure, the rate of heat loss from condenser for ORC decreases, whereas the rate of NPD by ORC increases. Therefore, the efficiency of ORC increases with increase in pressure. While that for DS-KC and BC with increase in pressure; the rate of heat loss from condenser increases, while the rate of power developed by cycles decreases after 72 bar and 52 bar, respectively. Therefore, efficiency of DS-KC and BC also deceases after a particular pressure value.

Figure 16 shows that with increase in pressure the installed cost per unit kW for DS-KC and ORC decreases. Because as it is evident that with increase in pressure the NPD by DS-KC and ORC increases keeping the installation as constant. But for BC, the ICPUK increases with increase in pressure. This happens only because the net power developed by BC decreases

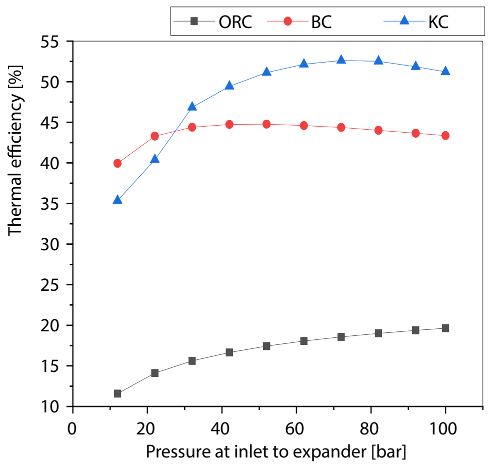

Figure 15. Variation of thermal efficiency *vs.* PIT

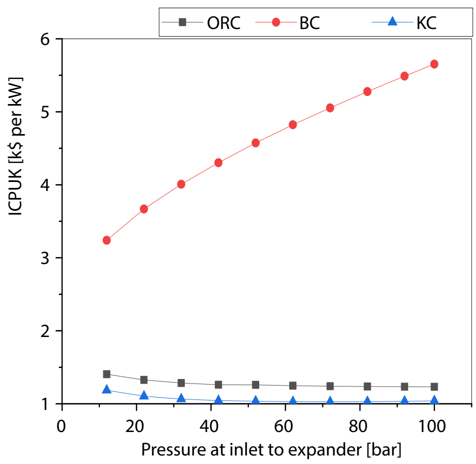

Figure 16. Variation of ICPUK vs PIT

after a certain pressure point, whereas the installation cost of compressor for BC also increases with increases in pressure.

Conclusions

The thermal efficiency DS-KC is greater than ORC and BC, if the number of PTC are less than 15. If the number of PTC are lesser than 5, then ORC has the highest efficiency. Also, it can be seen that if the number of PTC are beyond 15, then BC has the highest efficiency. It can be inferred that this sequence of ORC > DS-KC > BC is also economically viable with increase in number of PTC.

It can also be inferred that with increase in MFR from 0.5-1 kg/s, BC has the highest efficiency. Beyond a MFR of 1 kg/s the thermal efficiency of DS-KC has the highest value as compared to ORC and BC. Also, economically BC can be considered for the MFR lesser than 1 kg/s; while beyond 1 kg/s DS-KC and ORC can be considered. So, in this case the sequence of preference is BC > DS-KC > ORC with increase in MFR.

Also, with increase in pressure from 12-22 bar, the efficiency of BC is higher as compared to DS-KC and ORC. But, after a pressure value of 22 bar the efficiency of DS-KC becomes higher as compared to BC and ORC. Whereas from economic point of view DS-KC and ORC have lesser installation cost as compared to BC. Therefore, with increase in pressure; the sequence of preference with respect to thermal and economic considerations is DS-KC > ORC > BC.

Therefore, it can be seen that on an average; the thermal and economic performances of DS-KC is comparatively more as compared to BC and ORC.

Nomenclature

a – intermolecular force
A – heat exchanger area, [m^2]
Ar – aperture area, [m^2]
b – hard sphere volume
C – cost rate, [\$/h]
C_p – specific heat, [Jkg^{-1}K^{-1}]
h – specific enthalpy, [kJkg^{-1}]
HLC – heat lost by condenser, [kW]
I – solar intensity, [wm^{-2}]
$ICPUK$ – installation cost per unit kW, [k\$/kW]
m – mass-flow rate, [kgs^{-1}]
p – pressure, [bar]
P – power developed, [kW]
NPD – net power developed, [kW]
Q – net heat transfer, [kW]
$\dot{q}$ – heat flow, [kW]
T – temperature
$\dot{W}$ – power, [kW]

Greek symbols

η – efficiency
ρ – density, [kgm^{-3}]
ω – acentric factor

Acronyms

BC – Brayton cycle
DS-KC – double separator Kalina cycle
MFR – mass-flow rate
ORC – organic Rankine cycle
PTC – parabolic trough collector

Subscript

ab – absorbed
ap – aperture
bm – beam radiation
co – compressor
f – working fluid in the cycle
HE – heat exchanger
in – inlet
out – outlet
pr – produced
pu – pump
se – separator
tur – turbine

Superscripts

0 – ambient

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