The 6th International Symposium-Supercritical CO_2 Power Cycles,

March 27-29, 2018, Pittsburgh, PA

Feasibility Study of Supercritical CO_2 Rankine Cycle for Waste Heat Recovery

Ashish Chaudhary

Nabros Energy

Kheda (Gujarat), India.

Email: ne@nabrosenergy.in

Yagnesh Trivedi

Nabros Energy

Kheda (Gujarat), India.

Email: ne@nabrosenergy.in

Amit Mulchand

Nabros Energy

Kheda (Gujarat), India.

Email: ne@nabrosenergy.in

Haresh Chauhan

Nabros Energy

Kheda (Gujarat), India.

Email: ne@nabrosenergy.in

Punit Dave

Nabros Energy

Kheda (Gujarat), India.

Email: ne@nabrosenergy.in

Johnny Patel

Nabros Energy

Kheda (Gujarat), India.

Email: ne@nabrosenergy.in

Vishal Mehta

Nabros Energy

Kheda (Gujarat), India.

Email: ne@nabrosenergy.in

Ashish Chaudhary is a head of research team at the renewable energy division of Nabros Pharma Pvt Ltd, Kheda (Gujarat), India. He received a B.SC degree in chemistry from Vikram University of Ujjain, India in 1986. His areas of interest include the chemical & solvent extraction, distillation, design, analysis and optimization of energy conversion systems and the applications. He has been Head of Hydro thermal liquefaction team and has built up the first Hydro thermal liquefaction proto type plant in India. He has lead the team building the super critical carbon dioxide power generation cycle prototype plant in India.

Amit Mulchand is a founder director of the renewable energy division of Nabros Pharma Pvt Ltd, Kheda (Gujarat), India. He is a self-educated inventor having rich knowledge of conventional farming, marine farming with knowledge of two hundred different marine spices. His areas of interest include farming, chemical extraction, and distillation, power generation and the applications. He is the first person to develop Bowman-Birck Inhibitor, extract from soya protein with his own patented technology. Apart from these he is holding 22 different patents in various subjects. He is a pioneer investor of Hydro thermal liquefaction in India and has built up the first Hydro thermal liquefaction proto type plant in India. He is leading the team building super critical carbon dioxide power generation cycle prototype plant in India.

Yagnesh Trivedi is a researcher with formal qualification in Organisational Psychology. He has been leading the renewable energy division of Nabros Pharma Pvt Ltd, Kheda (Gujarat), India. His areas of interest include chemical extraction, energy conversion and the applications. He has been a core team member and head of the team, building up the first Hydro thermal liquefaction proto type plant in India & working on the possibilities of commercialisation of Hydro thermal liquefaction & super critical carbon dioxide power generation cycle projects.

Haresh Chauhan is a research assistant, assisting various research teams at the Renewable Energy Division Nabros Pharma Pvt Ltd, Kheda (Gujarat), India. He received a Commerce degree from Gujarat University of India in 1999. He has been working as facilities manager & Project coordinator of Hydro thermal liquefaction & Super Critical Carbon Dioxide Power Generation Cycle Team at Nabros Pharma Pvt Ltd since 2014. He has been a core team member building up the first Hydro thermal liquefaction proto type plant in India & working on the possibilities of commercialisation of Hydro thermal liquefaction & Super Critical Carbon Dioxide Power Generation Cycle projects.

Punit Dave is a research assistant at the renewable energy division Nabros Pharma Pvt Ltd, Kheda (Gujarat), India. He received MTech degree in thermal engineering from Pandit Dindayal Petroleum University of Gujarat, India in 2017. His area of interest includes thermo dynamic analysis of power plant and computational fluid dynamic analysis for turbo machinery and heat exchanger. He has been a member of the team, building super critical carbon dioxide power generation cycle prototype plant in India.

Johnny Patel is a research assistant & process engineer at the renewable energy division Nabros Pharma Pvt Ltd, Kheda (Gujarat), India. He received B.Tech degree in Mechanical Engineering from Univarsal College of Engineering and Technology affiliated to Gujarat Technological University of India in 2014. He has worked as an assistant engineer of India's first Hydro thermal liquefaction proto type plant. His area of interest includes thermo dynamic analysis of power plant and computational fluid dynamic analysis for turbo machinery and heat exchanger. He has been a member of the team, building super critical carbon dioxide power generation cycle prototype plant in India.

Vishal Mehta is a research assistant at the renewable energy division Nabros Pharma Pvt Ltd, Kheda (Gujarat), India. He received M.Tech degree in Thermal Engineering from Pandit Dindayal Petroleum University of Gujarat, India in 2017. His area of interest includes thermo dynamic analysis of power plant and computational fluid dynamic analysis for turbo machinery and heat exchanger. He has been a member of the team, building super critical carbon dioxide power generation cycle prototype plant in India.

Abstract

Supercritical carbon dioxide (S-CO_2) as a working fluid for power generation is currently thrust full research topic due to its high efficiency compare to steam Rankine cycle. Recuperation is one of the key component in S-CO_2 power cycle, but at the same time it puts a limitation to utilize complete heat source in waste heat recovery application. The range of medium waste heat source temperature is 500 K – 800 K. There is always high temperature of S-CO_2 fluid (ranging between 400 K to 600 K), after expansion from turbine. Hence there is always energy left unused in S-CO_2 at the turbine outlet. Going one step further, use of recuperator reduces the temperature difference between S-CO_2 and flue gas in main heat exchanger, which leads to ineffective heat exchange and continuous heat loss.

The present study analyzes the three cycles namely S-CO_2 Rankine cycle, steam Rankine cycle, S-CO_2 Rankine cycle combined with steam Rankine cycle as bottoming cycle, considering flue gas as a fluid of waste heat. The study compares the net power generation and loss of waste heat between these three cycles. We have considered 3.5% pressure drop between heat exchanger inlet and outlet. Result shows that loss of waste heat in S-CO_2 Rankine cycle is 40% while remaining both cycle has very negligible amount of waste heat loss. The net power generation is very low for S-CO_2 Rankine power cycle as compare to steam Rankine power cycle and combined power cycle, for the case of 1.18 MW reusable heat source. Also combined power cycle generates 10 % higher power than steam Rankine cycle.

Keywords

Waste Heat Recovery System, Supercritical CO_2 cycle, Rankine Cycle, combined power cycle.

1) Introduction

Waste heat is the heat which is generated in a process by way of fuel combustion or chemical reaction. This type of waste heat was dumped into the environment through it could still be reused for some useful and economic purpose. India has large quantity of hot flue gases generated from the Boilers, Kilns, Ovens and Furnaces. If some of its waste heat is recovered, considerable amount of primary fuel could be saved. The energy lost in the waste heat cannot be fully recovered. Waste heat can be classified from the temperature range: 1) High temperature Heat Recovery 2) Medium Temperature Heat Recovery 3) Low Temperature Heat Recovery [1]. Nickel, Aluminium, Zinc, Copper, Steel heat furnaces waste heat temperature range is 700-1000 ^{o}C which are included in the high temperature heat source. Steam, gas, boiler exhaust and Cement industries have medium temperature heat source.

The Indian Cement Industry with approximate 400 MiTPA (Million Ton Per Annum) is the second largest after China. In 2012, India has 146 cement plants which has production capacity of 346.2 MiTPA and per capita cement use is 191 kg [2]. The net heat load consumes by a cement clinker is 680 Kcal/Kg. In a cement plant, nearly 35% heat is lost primarily in preheater and cooler waste gases[3]. Steel industries and cement industries consumes 15.35% and 9.10% of total 164.97 Million toe [4] in India. The Waste Heat Recovery power plant can reduce the 70,000 tons of CO_2 emission per annum.

Japanese companies introduced first steam cycle from waste heat recovery in the cement industry which was installed by the Kawasaki Heavy Industries (KHI) at Sumitomo Osaka Cement. The first major commercial system with a capacity of 15 MW has been in operation since 1982. Initially, Waste Heat Recovery by China was driven by incentive tax breaks and Clean Development Mechanism (CDM) revenues for emission reduction from clean energy projects. Business opportunity revealed by the study that investment of US $ 5 billion to introduce ~2GW_e of waste heat recovery in eleven countries. Today, in India there are 22 Waste Heat Recovery projects commissioned from Indian Planning Commission. Vicat Sagar Cement industry held in Gulmarg plant (India), has WHR capacity of 8.4 MW and using steam cycle for power generation. JK Lakshmi cement- Nimbahera plant having production capacity of 4800 tpd (tonne per day) which can produce 12.1 MW net power generation. KCP Limited at Andhra Pradesh is producing 1600 tpd capacity and the net power generation capacity is 2.25 MW.

2) Objective

The objective of this study is to find best possible thermodynamic power cycle that suits the medium range waste heat recovery.

We have considered 1.18 MW of waste Heat source at 873 K temperature for three power cycle namely steam rankine cycle, S-CO_2 rankine cycle and combined cycle (supercritical CO_2 cycle as topping and steam cycle as bottoming). ORC (organic rankine cycle) cycle is suitable for low range heat source (up to 200 °C), above this temperature, Organic fluids are not stable.

We performed the first law and second law analysis of these three power cycles using Engineering equation solver to do comparison study.

3) Thermodynamic cycles

This section shows a description of the schematic diagram with state point of all three cycle. Also describe mathematical model and assumption that used for calculation in EES.

3.1 Thermodynamic analysis of Steam Rankine Power cycle

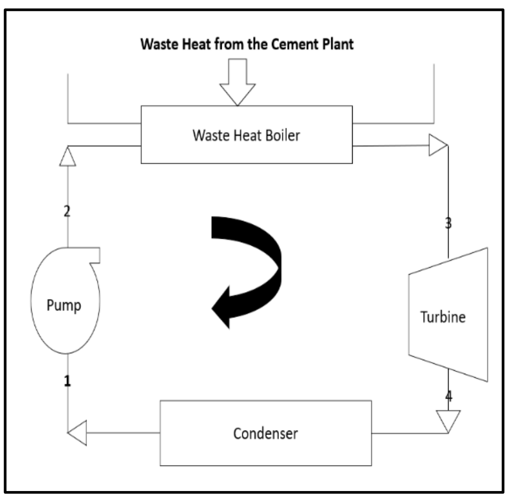

fig 1: -Schematic layout of Steam Rankine Cycle

Steam Rankine power cycle is very mature technology. Low temperature at the outlet of the turbine and high temperature of waste heat, water can utilize maximum energy from the waste heat. The phase change in the boiler, it required more latent heat. At the outlet of the primary heat exchanger, high temperature steam expands into the turbine.

Table 1: Transport properties at each state point

State	1	2	3	4
P, BAR	1	150	144.8	1.036
T, K	305	306.3	743.1	373.8
H, kJ/kg	133.5	152.2	3228	2268
S, kJ/(kg*K)	0.4619	0.4741	6.252	6.767

Assumption

I) Steam Turbine and pump both have 80% isentropic efficiencies [5].

$$\eta_t = \frac{(h_3 - h_{4a})}{(h_3 - h_4)} = 0.8 \tag{1}$$

$$\eta_c = \frac{(h_2 - h_1)}{(h_{2a} - h_1)} = 0.8 \tag{2}$$

II) A minimum temperature difference of 20 K is required at the between warm and cold streams in any heat-exchanger. Like, $T_{so} - T_{2a} = 20$

III) A pressure drops of 3.5% of entry pressure occurs in each stream during the heat addition and heat rejection processes.

Mathematical model for thermodynamic analysis

To determine the outlet temperature of working fluid in the boiler, we assume the same entropy generation at the both ends of boiler.

$$\left(dS_{gen}\right)_{si-3} = \left(dS_{gen}\right)_{so-2}$$

$$\left(dS_{gen}\right)_{si-3} = \left(\frac{T_{si} - T_3}{T_{si} * T_3}\right) * dQ$$

$$\left(dS_{gen}\right)_{so-2} = \left(\frac{T_{so} - T_2}{T_{so} * T_2}\right) * dQ$$

At the outlet of the waste heat exchanger, the temperature gain by the working fluid has been found by:

$$T_3 = \frac{(T_{si} T_{so} T_{2a})}{T_{2a} T_{so} + T_{si} T_{so} - T_{si} T_{2a}} \tag{3}$$

Where, T_{si} = Heat Source inlet temperature; T_{so} = Heat source outlet Temperature

Mass flow rate of steam can be find out by the energy balance equation.

$$m_{flue} C_{ps} (T_{si} - T_{so}) = m_{water} (h_3 - h_{2a}) \tag{4}$$

Power generated from the turbine in the combined cycle can be calculated as,

$$P_{ST} = m_{water} (h_3 - h_{4a}) \tag{5}$$

Heat Supplied by flue gas, $$Q_s = m_{flue} (h_{in} - h_{out}) \tag{6}$$

Exergy analysis component wise in compressor, turbine is given by the following equations:

$$I_{pump} = m_{water} T_0 (S_{2a} - S_1) \tag{7}$$

$$I_{turbine} = m_{water} T_0 (S_{4a} - S_3) \tag{8}$$

Exergy loss in the Heat recovery steam generation, condenser and primary heat exchanger is given by:

$$I_{BOILER} = T_0\big(m_{water}(S_3 - S_{2a}) - m_{flue}(S_{si} - S_{so})\big) \tag{9}$$

$$I_{condenser} = m_{water}(h_{4a} - h_1 - T_0(S_{4a} - S_1) \tag{10}$$

Performance Indicators

The parameters of interests are the overall thermal efficiency of cycle that defined as,

$$\eta_{th} = \frac{P_{ST} - P_{pump}}{Q_s} \tag{11}$$

Second law efficiency can be calculated as

$$\eta_{II} = \frac{Exergy_{supplied} - Exergy_{destroyed}}{Exergy_{supplied}} \tag{12}$$

$$Exergy_{supplied} = Q_s * \eta_{carnot} \tag{13}$$

Table 2: Overall performance Result

First Law efficiency			
(1)	Mass flow rate of Steam	kg/s	0.3662
(2)	Mass flow of flue gas	kg/s	2
(3)	Turbine Power Output	kW	281.4
(4)	Pump input power	kW	6.81
(5)	First law eff	%	24.38
Second Law efficiency			
(6)	Exergy Supplied	kW	739.2
(7)	I_turbine	kW	56.47
(8)	I_boiler	kW	19.4
(9)	I_condenser	kW	159.2
(10)	I_pump	kW	1.34
(11)	2nd law eff	%	68.02

3.2 S-CO_2 rankine cycle

Carbon dioxide can be used as the working fluid in the Supercritical CO_2 power cycle. S-CO2 rankine cycle is also called as transcritical CO_2 cycle. The critical point of CO_2 is 73.98 bar and 31^o C. The carbon dioxide operates between below and above the critical point in this cycle. Because of high pressure, the system becomes very compact as compare to steam cycle. CO_2 gained heat from the waste heat source and expands into the gas turbine. Because of low expansion ratio as compare to steam rankine cycle, at the outlet of the turbine has high temperature that can re-utilized from the recuperator.

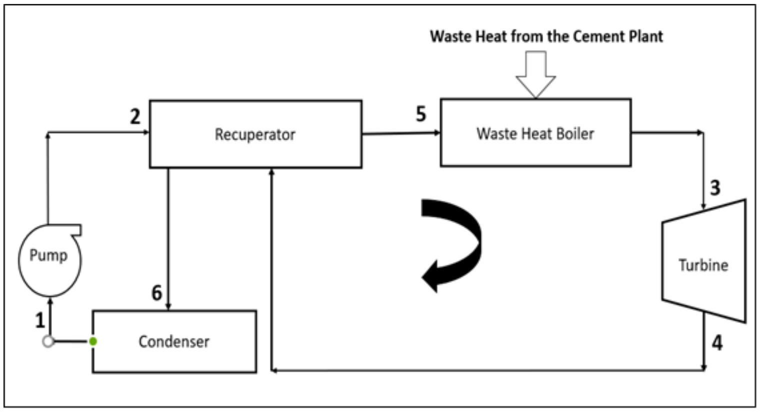

Fig 2: - Schematic Layout of Transcritical CO_2 Power Cycle

Table 3: Transport properties at each state point

State	1	2	3	4	5	6
P, bar	70	350	325.9	72.54	337.8	70
T, K	300	341.3	835	676.7	609.8	361.3
H, kJ/kg	-228.6	-184.4	535.8	365.8	246.6	5.341
S, kJ/(kg*K)	-1.482	-1.456	-0.0857	-0.021	-0.4967	-0.737

Thermodynamic analysis has been done using same mathematical model and assumption as described earlier for steam rankine power cycle.

Table 4: Overall performance

First Law efficiency			
(1)	Mass flow rate of CO_2	kg/s	1.732
(2)	Mass flow of flue gas	kg/s	2
(3)	Turbine Power Output	kW	294.4
(4)	Compress input power	kW	76.53
(5)	First law eff	%	18.46
Second Law efficiency			
(6)	Exergy Supplied	kW	774.7
(7)	I_turbine-destruction	kW	33.55
(8)	I_Primary hx - Destruct	kW	225.6
(9)	I_condenser-Destruct	kW	16.96
(10)	I_compressor	kW	13.55
(11)	I_recuperator	kW	127.6
(12)	2^{nd} law eff	%	46.14

3.3 Combined power Cycle

In the combined cycle, Supercritical CO_2 is the topping cycle and steam Rankine cycle as the bottoming cycle.

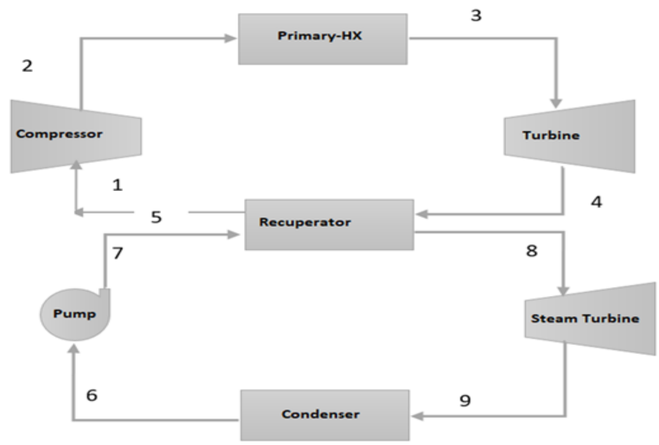

Fig 3: - Schematic layout of Combined Power Cycle

Table 5: Transport property at each state point

State	1	2	3	4	5	6	7	8	9
P, bar	70	350	337.8	72.54	70	1	45	43.43	1.036
T, K	300	341.3	764.7	610.2	325	305	305.4	544.4	373.8
H, kJ/kg	-228.6	-184.4	445.6	289.5	-49.32	133.5	139	2857	2290
S, kJ/(kg*K)	-1.482	-1.456	-0.2057	-0.1399	-0.8946	0.4619	0.4656	6.143	6.312

Thermodynamic analysis has been done using same mathematical model and assumption as described earlier for steam rankine power cycle.

Table 6: Overall performance of combined power cycle

First Law efficiency			
(1)	Mass flow rate of CO_2	kg/s	1.673
(2)	Mass flow of water	kg/s	0.2086
(3)	Mass flow of flue gas	kg/s	2
(4)	Gas Turbine Power Output	kW	261.1
(5)	Steam turbine power output	kW	118.4
(6)	Compressor input power	kW	73.91
(7)	Pump input power	kW	1.15
(8)	First law efficiency	%	28.88

Second law efficiency			
(9)	Exergy Supplied	kW	691.9
(10)	Exergy destroyed in CO_2 Turbine	kW	33.04
(11)	Exergy destroyed in Compressor	kW	13.04
(12)	Exergy destroyed in steam turbine	kW	10.56
(11)	Exergy destroyed in Primary HX	kW	72.44
(12)	Exergy destroyed in condenser	kW	83.69
(13)	Exergy destroyed in HRSG	kW	9.51
(14)	2nd law eff	%	67.87

4) Result and Discussions

First law and second law of thermodynamic analysis is carried out with the help of EES. There is optimum pressure ratio corresponding to the turbine inlet temperature in each cycle. Graph has been plotted for pressure ratio and net-work output from each system to do parametric study.

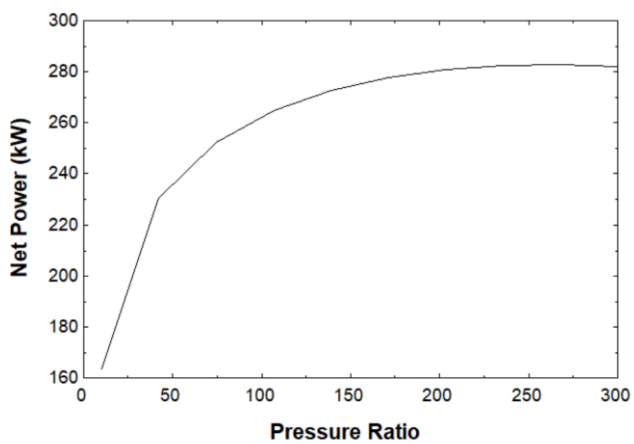

Figure 4: Graph between pressure vs Net power output for steam rankine cycle

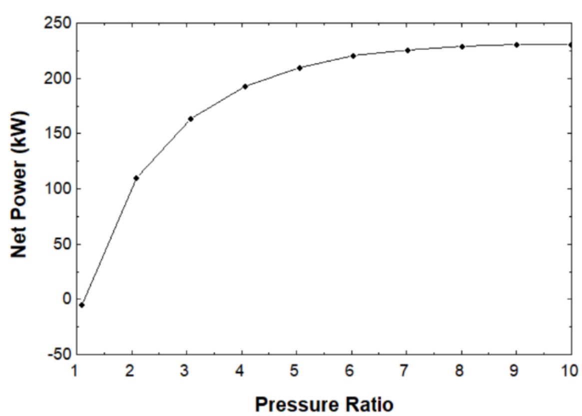

Figure 5: Graph between pressure vs Net power output for S-CO_2 rankine cycle

Optimum pressure for SRC is 150 bar because as shown in figure 4 above this pressure net power output changes becomes negligible. For S-CO_2 rankine cycle, optimum pressure ratio is 5 above this value changes in net power becomes negligible (shown in figure 5). For combined power cycle, optimum pressure ratio is 5 (shown in figure 6).

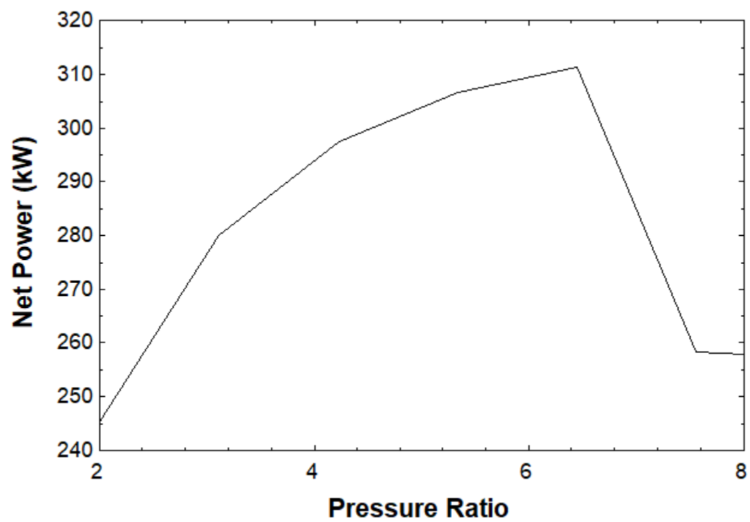

Figure 6: Graph between pressure vs Net power output for combined power cycle

Bar chart in figure 7 compares the first law and second law efficiency for these three power cycles. First law efficiency is maximum for combined cycle which is 28.9%. The main reason for maximum efficiency is no latent heat required leads to high temperature at turbine inlet as well as complete utilization of waste heat source by adding SRC in bottoming cycle. Exergetic efficiency is almost equal for combined power cycle and steam rankine cycle. T-CO_2 rankine cycle has very low exegetic efficiency.

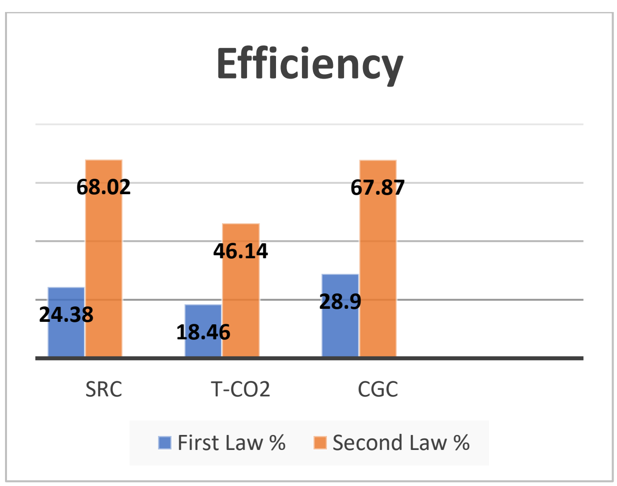

Fig 7: - First law and Second law efficiency for various power cycle

Irreversibility of condenser is very high as shown in figure 8, due to fact that all the energy has lost to the environment. Apart from condenser, turbine loss is high because of low isentropic efficiency.

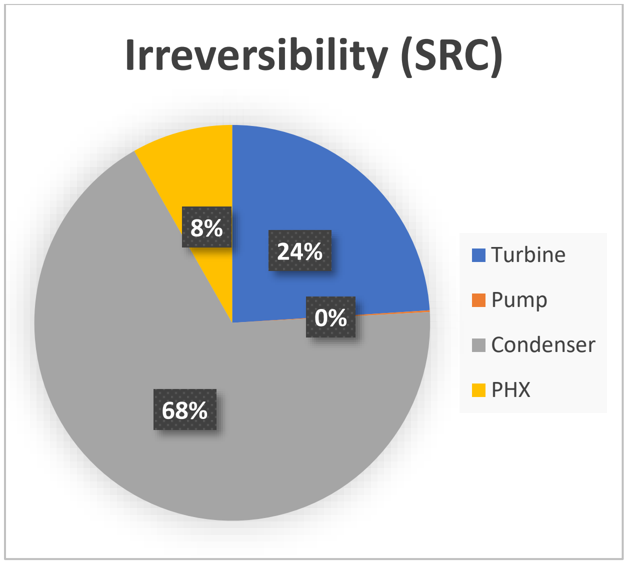

Fig 8: - Exergy Loss at each component in Steam Rankine Cycle

In T-CO_2 cycle, the maximum exergy destruction (shown in figure 9) is in the primary heat exchanger (PHE). Use of recuperator reduces the temperature difference between S-CO_2 and flue gas in main heat exchanger, which leads to ineffective heat exchange and continuous heat loss. The exergy destructed in turbine and compressor is because of its isentropic efficiency and its losses in the turbomachinery.

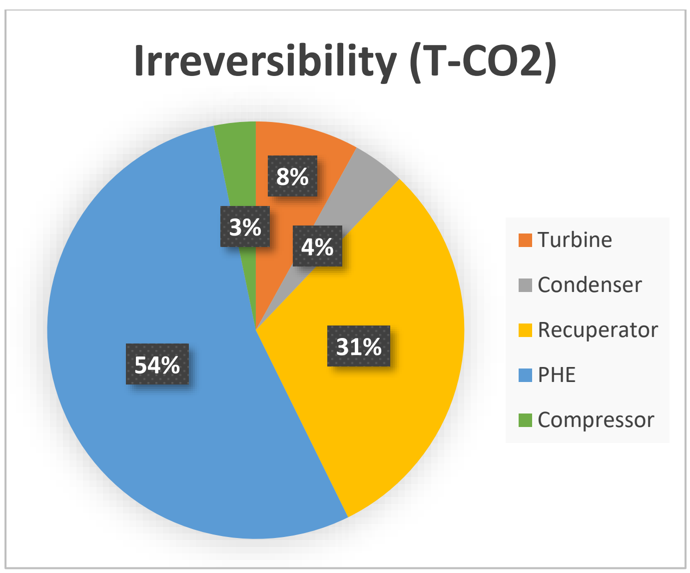

Fig 9: - Exergy Loss in each component in Transcritical CO_2 cycle

In the Combined power cycle, major part of exergy loss is in boiler and condenser. For that, the main reason is latent heat requirement in the boiler for the phase change. Second major irreversibility in the condenser because of phase change of steam. Exergy loss in the CO_2 turbine is higher as compare to steam turbine because of very high inlet temperature and law expansion ratio that increases the irreversibility. Steam turbine has higher expansion ratio but law temperature at the inlet of the turbine that reduces the irreversibility

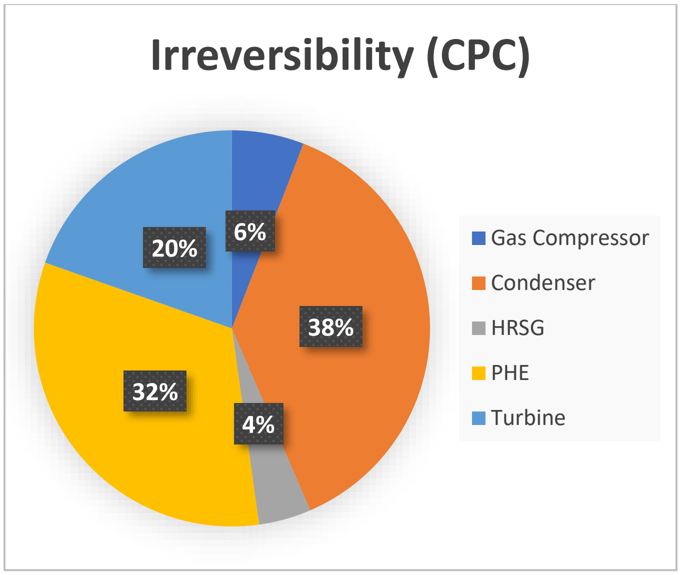

Fig 10: - Exergy Loss in each component in Combined Power Cycle

Steam Rankine cycle get highest exergy efficiency among the three power cycles. The highest exergy loss is in the condenser to phase change of the steam. Maximum exergy is destructed in the condenser and second major part is recuperator. The exergy loss in the recuperator for Combined cycle is less compare to S-CO_2 rankine cycle. Compressor and turbine has its isentropic efficiency that leads exergy loss. Pump exergy loss is very negligible as compare to other components due to its incompressible nature.

Conclusion

We discussed three power cycles namely Transcritical CO_2 cycle, Steam Rankine cycle and Combined Power cycle by using first law and second law thermodynamic analysis. From the first law analysis, result shows that combined power cycle is 28.9% efficiency is greater than S-CO_2 rankine cycle (18.46 %) and SRC (24.48 %) cycle for medium waste heat source.

The higher efficiency of combined cycle is because of carbon dioxide is above the critical point, so that there is no latent heat is required so it utilize maximum energy gain compare to others. The exergy efficiency is similar to SRC which is around 68%. S-CO_2 rankine cycle is not feasible for waste heat recovery because maximum exergy destroyed during the heat exchange and exergy efficiency is 46%.

References

1) Waste heat recovery report from India

2) http://www.iipnetwork.org/62730%20WRH_Report.pdf2

3) http://www.holtecnet.com/holtecdocs/TechnicalPapers/p_2011_6.pdf

4) http://www.holtecnet.com/holtecdocs/TechnicalPapers/p_2017_1.pdf

5) “Supercritical carbon dioxide Brayton cycle for concentrated solar power”, the journal of supercritical fluids, Pradeep Garg, Pramod Kumar, Kandadai Srinivasan, Science Direct jan (2013)