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Performance evaluation of solar parabolic trough collector powered super critical CO_2 cycle intercooled by organic Rankine cycle using ecofriendly working fluids

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Abstract

In this work, supercritical carbon dioxide with double compressor cycle is considered in which solar parabolic through collectors are taken for heat source. The ORC bottoming cycle is currently incorporated into the exhaust of recuperative gas turbines to further lower the temperature of the exhaust gas, yielding similar overall efficiency to that of conventional gas turbine and steam combined cycles. However, a certain amount of thermal energy in the intercooler is not effectively utilized in the intercooled gas turbine cogeneration cycles. The temperature of the compressed SCO_2 at the intercooler inlet could be found about 120^0C -150 ^0C. This is an ideal energy source to be used in an ORC for power generation. In this investigation, a thermodynamic analysis was carried out on overall cycle (SPTC-SCO_2-ORC) combined cycles comprising intercooled to recover waste heat from the intercooler of SCO_2 gas turbine. Two existing gas turbines were performed as the topping cycles with appropriate modifications. The following organic fluids were considered as the working fluids in ORCs: R134a, R245fa, R12345yf, R236fa, R227ea, benzene, isobutene, isopentane, cyclohexane, and R410a. A computer program was then designed for computations of system performance. Thermodynamic analyses were performed to study the effects of parameters including solar irradiation, velocity of HTF in absorber tube and pressure at the ORC turbine inlet on the performance on combined cycle. From the study it is found that increase in solar irradiation overall performance of system is improved such as first law and second law efficiencies and net-work output. In contrast the exergy destruction ratio is reduced i.e. irreversibility are reduced. It has been found that SPTC is the primary source of exergy destruction in which more than 79.11% of the solar inlet exergy has been destructed in the solar collector field only, which is a crucial amount. Therefore, it requires a necessary care during the designing of SPTC plant to decrease down the exergy destruction rate in the solar driven combined cycle.

Keywords: Thermodynamic analysis, SCO2 cycle, intercooling, ORC, waste heat recovery

1. Introduction

Different heat source such as coal power, natural gas, high temperature fuel cell, solar thermal energy can be used effectively by Supercritical CO2 cycle [1]. Harvesting the solar energy became important now a day because the it is clean and green and pollution free so it can be more effective in near future. So near future utilization of solar energy will be upgraded the performance of thermal power plan. There. are many solar collector technologies which are being used for harvesting solar energy, but parabolic solar through collectors is being used as the most important technology for power generation. Thermodynamics analysis point of view solar parabolic through collector's system is considered efficient heat source. Nowadays, supercritical CO2 cycle and organic Rankine cycle integrated with various renewable heat sources are considered for the purpose of power generation [2–8]. Cheng Zhou [2] compared analyzed different super critical ORC cycle with the different sub critical hybrid cycle driven with the geothermal and solar combined and also standalone. Finally, he concluded that the super critical cycle shows better performance the sub critical cycle driven same heat source and electricity efficiency is exceeding 5-16%. Jing Li et al. [3] proposed a model of solar integrated ORC system using direct vapor generation system and find the performance by using 17 dry and isentropic working fluids. They concluded that the efficiency of combined ORC and

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the direct vapor generation system increases with increase in critical temperature of fluids and also with the collector efficiency reduces continuously. They further concluded that R123 is best fluid performance point of view among other selected working fluids. Al-Sulaiman [4] proposed a solar parabolic through collectors integrated model of combined organic Rankine cycle and steam Rankine cycle and found that R134a gives better performance among other selected working fluids. further found that R134a gives better second law efficiency almost 26% which is highest among the other selected working fluids. Niu et al. [5] proposed a model of solar integrated super critical Rankine cycle with deferments arrangements such as five units in series, parallel and cascade. Then finally concluded that solar parabolic through collectors integrated with cascade system gives maximum work output. Cardemil et al. [6] carried out a thermodynamic model of supercritical CO2 integrated Rankine and Brayton power cycles using deferent's working fluids such as ethane, D4siloxane, toluene and water then performance analysis is done. He concluded that first law efficiency of system integrated with co2 is lower than the other working fluids while second law efficiency higher than the selected working fluids. Garga et al. [7] conducted a study in which he compared the transcritical CO_2 cycle (i.e high pressure and temperature) with the transcritical steam Rankine cycle. They concluded that variations of temperature did not affect the performance of transcritical CO2 cycle and also further found that this transcritical cycle requires only single heat transfer fluid (HTF) loop as compared to trans-critical steam cycle coupled with two HTF (heat transfer fluids) loops in series. Osorio et al. [8] proposed a model to find the dynamic behavior of SCO2 power cycle integrated with a concentrated solar power system (i.e. central receiver), heat exchange device, recuperator, hot and cold energy storage and multi-stage compression-expansion subsystems along with the reheater and intercooler as an integral component employed between the turbine and compressor. Their results showed that the maximum power output and process efficiency and is1.6 MW and 21% respectively. At last, they concluded that the SCO2 cycle's operating time after optimization was increased from 220 to 480 min because of thermal storage application Further, few researchers considered the ORC integrated with SPTC with various applications like waste heat recovery and cogeneration process. Nafey and Sharaf [9] conducted a study for thermodynamic performance analysis and cost evaluation of the ORC using (SPTC) solar parabolic trough collector as a heat source for generating mechanical power for driving desalination system by using reverse osmosis (RO). Delgado-Torres and García-Rodríguez [10,11,12] performed a detailed thermodynamic analysis of ORC integrated with the parabolic trough collectors and a ORC system for seawater RO unit for production of water by RO (reverse osmosis) process [10], and in another study, they performed experiments for investigations related to preliminary designs of the low-temperature solar thermal collector driven RO desalination for sea water and brackish water. It concluded that by using R245fa in a solar thermal integrated RO system, the production of solar desalination system could be increased up to a maximum value exchanger area, and with the help of genetic algorithm a set of optimal solution obtained. Finally, they found the solution of (i.e. below 2%) [11], further they also performed a study to investigate the effect of different working fluids such as R245ca, butane, isobutene, and R245fa on the aperture area of the SPTC system for water desalination and power production [12]. Al-Sulaiman et al. [13] performed a study to assess the performance of a novel system integrated with SPTC and ORC for combined cooling, heating and power (CCHP). Further They used a fraction of waste heat for ORC for heating as well as cooling cogeneration and also investigate the different output parameters. Finally, his study reveals that the electrical efficiency significantly improved from 15% to 94% for a solar mode (i.e. without energy storage). Gao et al. [14] proposed a model and conducted a performance analysis of solar driven ORC and concluded that the efficiency increases as the inlet pressure and temperature of turbine increases when the system is located above the critical temperature limit. Wang and Dai [15] conducted an exergo-economic and comparative study of SCO2/tCO2 and SCO2/ORC configuration and concluded that at a lower compression pressure ratio, the SCO2/tCO2 cycle performs better than SCO2/ ORC. Moreover, it was investigated that that as compared to SCO2/tCO2 cycle, the SCO2/ORC cycle has slightly more economic. Singh and Mishra [16] carried out a performance analysis of the SPTC integrated supercritical ORC and found that R600a possess the maximum value of exergy efficiency which is around 96.09% at direct solar irradiation of 0.95 kW/m^2. At last, their study shows that fuel depletion ratio, improvement potential, and irreversibility ratio in case of SPTC system was found to be 11859 kW, 0.579 and 0.9296, respectively. Ferrara at al. [17] performed a thermodynamic analysis of ORC integrated with concentrated solar power system and found from their optimization analysis that acetone is the best working fluid for ORC system as compared to R245fa and R134a. Calise et al. [18] proposed a model to investigate the performance of the system based on evacuated flat-plate collector and ORC under the different climate conditions. It was investigated that the efficiency of ORC always low close to 10% during the whole year as compared to solar collector whose efficiency was high (>50%) in summer and low (down to 20%) in winter. Rayegan and Tao [19] developed a model to compare the capability of deferent working fluids such as refrigerant and non-refrigerant used for a solar Rankine cycle. They found that the refrigerants (R-245ca and R-245fa), high performance non-refrigerants (Acetone and Benzene) and medium performance non-refrigerants (Butane, Cis-butene, Isopentane, and Trans-butene) can be effectively utilized in a solar ORC at medium temperature level. Lastly, their results showed that as the collector efficiency increases from 70% to 100%, exergy efficiency and the enhancement in the limit of irreversibility reduction was reported around 5% and 35%, respectively. Hettiarachchi et al. [20] conducted an optimization study of low temperature ORC and geothermal as heat source. Found that ammonia uses the maximum amount of geothermal water. It was also observed that ammonia is a best choice as working fluid according to the ratio of efficiency to objective function as compared to n-Pentane, HCFC 123, and PF5050. Gimelli et al. [21] performed a multi objective model of the ORC system to maximize the efficiency as well as overall heat

optimal with the range of electrical efficiency (i.e. 14.1% to 18.9%) and heat transfer area (i.e. 446–1079 m^2). ORC as well as

CO_2 power cycle can be utilized for the exhaust/waste heat recovery process. Many Firms such as Echogen -power systems LLC (Ohio, USA) and General Electric (New York,) have already patents relevant to this application [22,23]. It is obvious to understand from the literature review that there is no thermodynamic analysis (energy and exergy) analysis of combined SCO2 and ORC cycle integrated with SPTC. In the literature, exergy analysis of the combined ORC and steam Rankine cycle was performed [4]. This paper focuses its attention on the combination of SCO2 and ORC (as intercooler cycle), because SCO2 cycle is able to replace the steam Rankine cycle due to some reasons for such it is less corrosive then steam Rankine cycle instance at the same which can effectively increase the inlet temperature of the turbine [1]. Also, smaller cycle pressure ratio and high outlet temperature of the turbine has been noticed in case of SCO2 cycle as compared to steam Rankine cycle which results in increases the thermal efficiency [1]. Therefore, the concept of combined configuration in the current study is original and its aim is to find an energetic and exergetic performance of the combined CO2 and ORC power system integrated with SPTCs plant. Based on this, exergetic performance parameters like the rate of exergy destruction ratio, overall first law and second law efficiency, overall work output was also examined in this research work.

2. System description

The present study considers a combined system integrated with solar parabolic trough collector as shown in Fig.1. In this configuration, the SCO2 cycle is a topping cycle while the ORC is a bottoming cycle or as intercooler cycle and it is directly equipped with SCO2 cycle for the purpose of utilization of waste heat which is wasting because of intercooling. CO2 as a working fluid used in the topping cycle at the critical conditions (i.e. 30.98 ^{0}C and 7.38 MPa) and it becomes incompressible near the critical point [1]. In the literature, the SCO2 density has been compared with the water density by Wright et al. [43] and it has been found that the density of CO_2 is 60.01% of the water density which is the inlet of compressor and it can be effectively reduced the need of compression power [41]. Fig.3 illustrates the variations of density of carbon dioxide (CO_2) at different temperature and pressure conditions and it is found that the density is very high density around the critical point, so in this way compression work reduced considerably as compared to other fluids [41]. it has been also being found that thermal conductivity of CO_2 possesses the highest value at the critical point which is around 148.95 mW/ m-K at 30.98 ^{0}C [41]. It is also found that the specific heat of CO2 varies radically with temperature and pressure variations is shown in Fig.2.

It is found that the difference in temperature of fluids fluctuates broadly with in recuperator which directly affects the design of recuperator with respect to the pitch point location [41,49,50]. Moreover, combined cycle also has an advantage over simple configuration because it can reduce the system design complexity due to condensation at atmospheric pressure not upon vacuum pressure as in simple configuration [4,51]. Multiple solar thermal collectors (i.e. SPTC field) are considered as a heat source for the combined cycle. SPTC field consists of 50 modules that are arranged in series per collector row and each having length of 12.27 m [4,13,24] and efficient single axis tracking system can be employed to track the sun movement so as to maximize the efficiency. Moreover, solar loop can be equipped with thermal storage facility to avoid the situation of sun set or blocked by clouds but this facility also includes some type of costs such as operating, storage medium, piping, containers and insulating materials which results in increases the operational cost of overall plant. Table 3 lists the data related to geometrical parameters selected for a solar collector and Table 2 lists the thermal properties of working fluid flowing through the collector. Some refrigerants were selected for investigating the low-temperature ORC based on literature review. For specific heat source conditions.

Table 1: Properties of organic working fluids. [38,39,47]

Fluids	Weights (kg/mole)	Critical temp (^{0}C)	Critical pressure(MPa)	Life (years)	ODP	GWP
R134a	102.03	101	4.059	14	0	1430
R1234yf	114.04	94.7	4.597	na	0	<1
R245fa	134.05	154.1	3.65	7.7	0	1050
R236fa	152.04	124.92	3.2	10	0	9810
R227ea	170	101.7	2.9	na	0	3220
isobutane	58.1	134.7	3.63	0.016	0	20
cyclohexane	84.16	280.5	4.075	na	na	na
isopentane	72.1	187.2	3.38	0.009	0	20
benzene	78.11	289	4.89	na	0	<2.6
R410a	72.6	72.6	4.86	12	0	1725

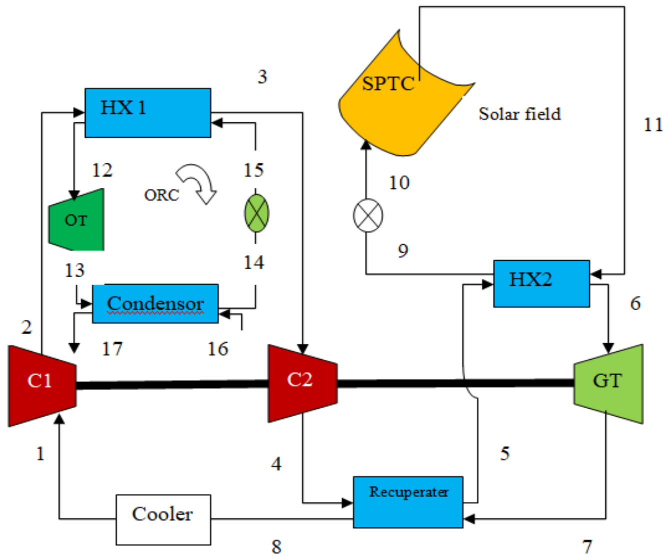

Figure 1: Schematic of the solar parabolic trough collectors integrated with SCO2 cycle and ORC as intercooler.

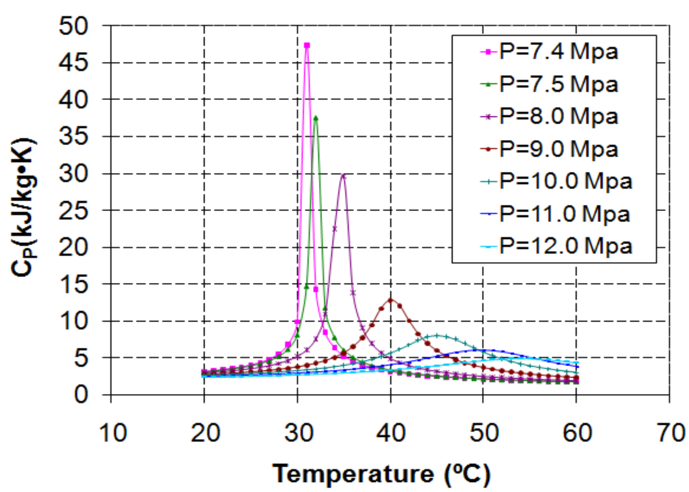

Figure 2: Variation of specific heat of super critical CO2 with temperature at different pressures [40]

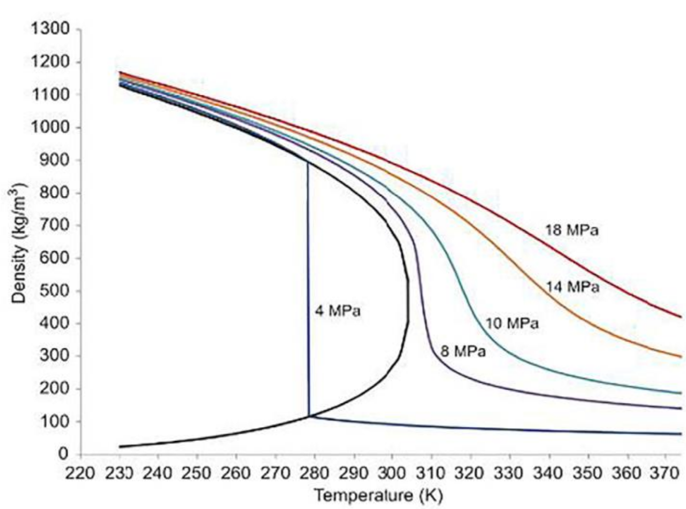

Figure 3: Variation density of CO2 with different temperature and pressure [41]

Refrigerants are selected: R134a, R410a, R1234yf, R236fa, R227ea isobutene, cyclohexane, benzene, isopentane and R245fa as listed in Table 1. These refrigerants are well suited for the low-temperature ORC coupled in combined cycle as a bottoming cycle and the fluid selection process is depending upon the thermodynamic and heat transfer properties, environmental as well as economy aspects [45,48.49]. Syltherm 800 is selected as the heat transfer fluid (HTF) for solar collector due to its maximum working temperature range of 420 ^{0}C [25] and it is best suited in this application amongst the other types of working fluids. In addition, it has a mass flow rate of 0.575 kg/s and 100 bar operating pressure in the SPTC field. The modified LS-3 (Luz third generation trough collector) is a latest SPTC design which has been chosen from the solar electric generating system with the collector row exit temperature of 400 ^{0}C (i.e. 673 K) [26]. The reason to choose LS-3 collector is that it has a larger aperture than LS-2 which results in 15% more receivers are required in case of LS-2 field. Also LS-3 has a lower mirror cost per square meter as compared to LS-2 collector [44]. In the combined cycle, SCO2 at high pressure and temperature expands in the turbine (process 6 to 7) up to a low temperature and pressure and then it goes to recuperator (process 7 to 8), where it extracts thermal energy from the hot stream and utilize this part of the energy to preheat the cold stream. After recuperation, stream - cooled in the cooler unit (process 8 to 1) and then it goes to compressor unit which is driven by turbine (process 1 to 2), where stream pressure and temperature increases again. After this the stream goes to heat exchanger (process 2 to 3), where stream gives the sufficient energy input to the ORC. In condenser water is used for cooling purpose (process 16 to 17). Then stream after passed through the heat exchanger 1 it reaches to compressor 2 for recompression (process 3 to 4) Then passing through the recuperatore stream goes to heat exchanger 2 unit (process 5 to 6) where SCO2 steam extracts heat from the syltherm 800 fluid flowing through the SPTC system.

Table 2: Thermal properties of syltherm 800 at 650K [28]

Thermal properties	value
Specific heat capacity (c_p)	2218.26 [J/kg K]
Density (ρ)	577.70 kg/m^3
Thermal conductivity (k)	0.067833 [W/m K]
Viscosity (μ)	0.000284 Pa-s

3. Mathematical modeling

The modeling of combined cogeneration system is discussed in this section. The mathematical modeling of large-scale solar parabolic trough collector (SPTC) has been performed which is followed by the modeling of combined SCO2 and ORC system. Exergy analysis of the SPTC system has been conducted by using the equations derived from previous research [4,29,30, 47] and solved by computational numerical technique, i.e. Engineering Equation solver (EES) Software. Apart from this, assumptions made that the pressure drop in the system is neglected except in case of pump and turbine; the system is at steady state which mean that system should be in unchanged condition even after transformation; pump and turbine efficiency are always constant as mentioned in Table 3 for all the organic fluids.

Table 3. Input data for the simulation

Solar parabolic through collectors[4,26]	
Collector row length	500 m
Collector type	Modified LS-3
Collector width	5.76 m
Collector length(single)	12.27m
Inner diameter of absorber tube	0.05m
Outer diameter of absorber tube	0.07m
Inner diameter of cover	0.115m
Outer diameter of cover	0.121m
Emittance of cover	0.86
Emittance of absorber tube	0.15
Reflectance of mirror	0.94
Intercept factor	0.93
Transmittance of glass cover	0.96
Absorbance of absorber tube	0.96
Shading loss	0.97
Structural loss	0.95
Concentration ratio	82:1
Intensity of direct radiation (w/m^2)	500-950
Incidence angle modifier	1
No. of collectors in series (cols)	50 [13]
No. of parallel collectors rows(colp)	7 [13]
Row orientation	North-south
Mirror optical efficiency	73.27%
Maximum outlet pressure	100 bar
Maximum outlet temperature	673 K
Ambient conditions	
Ambient temperature	298.15K
Ambient pressure	101.3 Kpa
CO2 cycle configuration	
Gas turbine efficieny	90% [27,44]
Both compressors efficiency	89%[27,28]
Organic pump efficiency	85%[27,44]
Organic turbine efficiency	87%[27,44]
Mass flow rate of CO_2	2 kg/s
Pinch point temperature	5℃ [27,44]
SCO2 cycle high pressure [27,42,44]	25 MPa
ORC turbine inlet pressure	4 MPa
Mass flow rate of ORC	1 kg/s
Recuperatore effectiveness	0.95[27,44]
Heat exchanger effectiveness	0.92

3.1 Exergy model for SPTC integrated with ORC

In this section, modeling of the SPTC based combined cycle integrated with ORC is discussed. In the literature, Al-Sulaiman [4] conducted the modeling of the parabolic trough collector with receiver tube along with the evaluation of exergy destruction ratio, exergy efficiency (second law), and energy efficiency (first law) exergetic fuel depletion ratio for combined cycle. Useful energy collected by the solar collector per unit time is defined as:

$$\dot{Q}_u = \dot{m}_a * Cp_a * (T_{a_o} - T_{a_i})$$

where C_p is the specific heat and m_a is the mass flow rate of liquid flowing in the absorber tube. The subscripts a_o and a_i refers to the absorber outlet and absorber inlet. Further, the useful heat gain can also be evaluated from the another formula which is given by:

$$\dot{Q}_u = A_p \,.\, F_R \,.\left(S - \frac{A_a}{A_p} U_L (T_{ai} - T_0)\right)$$

where F_R is a collector heat removal factor, A_p is the area of aperture, S is the heat flux absorbed by absorber tube, T_0 is the atmospheric temperature, and $U_L$is the overall heat loss coefficient of the solar collector. Absorbed heat flux and aperture area can be defined as

$$S = \eta_a * I_b$$
$$\eta_a = \rho_r * \alpha * \gamma * \tau * K_m$$
$$A_p = (W - D_{co,o})$$

Where W is the width of collector, ηa is the efficiency of absorber or receiver, $D_{co,o}$ is the outside diameter of cover, L is the collector length, ρ_r is mirror's reflectance, α is absorbance of absorber tube, γ is intercept factor, τ is glass cover's transmittance, Km is incident angle modifier which is determined by dividing the instantaneous thermal efficiency (ηi) at a given value of angle of incidence to the peak efficiency of SPTC [32], and I_b is direct solar irradiance. Apart from these parameters, the ratio of S to I_b gives the efficiency of the absorber tube η_a. All the necessary data related to these parameters has been listed in Table 3. Further, heat removal factor is defined as below

$$F_R = \frac{\dot{m} * C_{Pa}}{A_a * U_L}(1 - \exp\left(-\frac{F * A_a * U_L}{\dot{m} * C_{Pa}}\right))$$

where $A_a = \pi * Do * L$ and F is collector efficiency factor which is defined as below:

$$F = \frac{U_o}{U_L}$$

where U_o is the overall heat loss coefficient between surrounding and fluid flowing through the absorber tube and U_L is the heat loss coefficient of solar collector between ambient and absorber tube which is defined as:

$$U_o = \left[\frac{1}{U_L} + \frac{D_{a,o}}{h_{coa,i} \,.\, D_{a,i}} + \left(\frac{D_{a,o}}{2K_a}\ln\left(\frac{D_{a,o}}{D_{a,i}}\right)\right)\right]^{-1}$$

Where $h_{coa,i}$ is the heat loss coefficient between absorber and glass cover as shown below:

$$h_{coa,i} = \frac{Nu_a * k_a}{D_{a,i}}$$

Now heat loss coefficient of solar collector is defined as below:

$$U_L = \left[\frac{A_a}{(h_{c,amco} + h_{a,amco})A_{co}} + \frac{1}{h_{r,coa}}\right]$$

where subscripts a & co refers to the absorber and cover, $h_{c.amco}$

is the convection heat loss coefficient between ambient and cover, $h_{r,amco}$ is the radiation heat loss coefficient, and $h_{r,coa}$ is the radiation heat loss coefficient between absorber and glass cover which is defined as below:

$$h_{c,amco} = \frac{Nu.K_{air}}{D_{co,o}}$$

$$h_{r,amco} =$$

$$h_{r,amco} = \varepsilon_{co}.\sigma.({T_{\infty}}^2 + {T_{am}}^2)(T_{co} + T_{am})$$

$$h_{r,coa} = \frac{\sigma.(T_{co} + T_{a,avg})({T_{co}}^2 + {T_{a,avg}}^2)}{\frac{1}{\varepsilon_a} + \frac{A_a}{A_{co}}\left(\frac{1}{\varepsilon_{co}} - 1\right)}$$

Here the subscript *am* refers to ambient and *avg* refers to average, K_{air} is thermal conductivity of air, Nu is Nusselt number, σ is Stefan–Boltzmann constant, *eco* is emittance of the cover, *ea* is emittance of the absorber. Further, temperature of cover is defined as:

$$T_{co} = \frac{h_{r,coa}T_{a,am} + \frac{A_{co}}{A_a}(h_{c,amco} + h_{r,amco})T_{am}}{h_{r,coa} + \frac{A_{co}}{A_a}(h_{c,amco} + h_{r,amco})}$$

The total amount of solar flux directed (i.e. beam irradiation) upon the SPTC which is assumed as total heat available for the combined system.

$$\dot{Q}_s = A_p.F_R.S.Col_s.Col_p$$

where Col_s & Col_p is the total no. of collectors per single row in series and the total no. of collectors in parallel rows arrangement. Energy efficiency of SPTC can be expressed as [46]:

$$\eta_{en,SPTC} = \eta_o - c_1\frac{(T_m - T_{a,})}{I_b} - c_2\frac{(T_m - T_a)^2}{I_b}$$

where η_o is the optical efficiency the SPTC, c_1 is the first order coefficient$[W/m^2\,°C]$ and c_2 is the second order coefficient$[W/m^2\,°C]$, T_m is the mean temperature of heat transfer oil which is defined as: $T_m = \frac{T_{11}+T_9}{2}$

Velocity of HTF in absorber tube is given as:

$$V = \frac{4 * \dot{m}_{HTF}}{\pi * {D_i}^2 * \rho_{HTF}}$$

Where $\dot{m}_{HTF}$ is heat transfer fluid mass flow rate in solar field, D_i is inner diameter of absorber tube in solar field, ρ_{HTF} is density heat transfer fluid flowing in solar field.

Further, the term exergy is defined as the theoretical maximum work obtained from system as it interacts with the surrounding in an equilibrium condition. Therefore, steady state exergy balance of the control volume for each component based on physical boundary approach in a combined cycle is defined as:

$$\sum\left(1 - \frac{T_0}{T_Q}\right)\dot{Q}_Q - \dot{W}_{c.v} - \sum(\dot{m}_i Ex_i) - \sum(\dot{m}_e Ex_e) - \dot{Ex}_d = 0$$

Where $\dot{Ex}_d$ is the rate of exergy destruction, subscripts 0 and Q refers to the value of physical property at surrounding or dead state (i.e. T_0 = 298.15 K and P_0 = 101.3 kPa) and for a particular state, subscripts e and i refers to the exit and inlet state. Further, the Ex is the exergy per unit mass flow rate and chemical exergy value assumed to be negligible in the system. Exergy destruction in cooler also neglected. Now physical exergy per unit mass flow rate after neglecting the change in both velocity and elevation can be defined as under [4, 33]:

$$Ex^{ph} = (h - h_0) - T_0(s - s_0)$$

where h and s are specific enthalpy and specific entropy. Further, exergy at the inlet point of system (Ex_{inl}) is known as the maximum useful work available from solar radiation which is calculated by the Petela's formula as defined below [4,34,35,36]

$$Ex_{inl} = A_p * I_b * \left[1 + \frac{1}{3}\left(\frac{T_0}{T_{su}}\right)^4 - \frac{4}{3}\left(\frac{T_0}{T_{su}}\right)\right]$$

where, T_{su} is the temperature of superficial surface of the sun (black body) i.e. 5800 K [4]. Exergy (E_{xu}) gain by working fluid from the SPTC can be expressed as [46]:

$$\dot{Ex}_u = \frac{Q_u}{T_3 - T_1}\left[(T_3 - T_1) - \left(T_0 \ln\frac{T_3}{T_1}\right)\right]$$

The exergy efficiency (second law) of SPTC can be defined as:

$$\eta_{ex,SPTC} = \frac{\dot{Ex}_u}{Ex_{inl}}$$

Further, the input parameters required for the exergy and energy analysis of overall cycle has been listed in Table 3. Also, modeling of the overall cycle is based on the thermodynamic as well as exergetic equations which are derived from the literature [37]. The overall exergy efficiency (second law) can be defined as the ratio of net electrical output to the exergy at input. $\eta_{second\ law}$ for overall cycle is given by:

$$\eta_{second\ law} = 1 - \frac{Ed_{with\ PTSC}}{Ex_{inl}}$$

Where $Ed_{with\ PTSC}$ is total exergy destruction of plant considering parabolic through solar collectors and it is define as

$$Ed_{with\ PTSC} = Ed_{compressore\ 1} + Ed_{heat\ exchanger\ 1} + Ed_{compressore\ 2} + Ed_{heat\ exchanger\ 2} + Ed_{heat\ exchanger\ 3} +$$

$Ed_{gas\ turbine} + Ed_{Organic\ turbine} + Ed_{condensore} + Ed_{organic\ pump} + Ed_{PTSC}$

Neglecting exergy destruction solar field pump and cooler.
Now the fraction of total exergy destruction of a component can be defined as the difference of input and output exergy, which is calculated as:

$$Ed_{compressore\ 1} = (Ex_2 - Ex_1) + W_{compressore\ 1}$$
$$Ed_{heat\ exchanger\ 1} = (Ex_3 - Ex_2) + (Ex_{12} - Ex_{15})$$
$$Ed_{compressore\ 2} = (Ex_4 - Ex_3) + W_{compressore\ 2}$$
$$Ed_{Recuperator} = (Ex_5 - Ex_4) + (Ex_8 - Ex_7)$$
$$Ed_{heat\ exchanger\ 3} = (Ex_6 - Ex_5) + (Ex_9 - Ex_{11})$$
$$Ed_{gas\ turbine} = (Ex_7 - Ex_6) - W_{gas\ turbine}$$
$$Ed_{Organic\ turbine} = (Ex_{13} - Ex_{12}) - W_{organic\ turbine}$$
$$Ed_{condensore} == (Ex_{14} - Ex_{13}) + (Ex_{17} - Ex_{16})$$
$$Ed_{organic\ pump} = (Ex_{15} - Ex_{14}) + W_{organic\ pump}$$
$$Ed_{SPTC} = (Ex_{inl} - Ex_u)$$

Total heat input provided by solar collector to the combined cycle in evaporator unit can be defined as

$$\dot{Q}_{inl} = \dot{m}_{SCO2} * (h_6 - h_5)$$

where $\dot{m}_{SCO2}$ is the mass flow rate of SCO2 in the topping cycle. Thermodynamic process (6 to 7) in SCO2 turbine can be described as:

$$W_{gas\ turbine} = \dot{m}_{SCO2} * (h_6 - h_{7s}) * \eta_{gas\ turbine}$$

where h_{7s} is the isentropic enthalpy at the outlet of SCO2 turbine. Thermodynamic balance in the recuperator (process 4 to 5) can be expressed as:

$$\dot{Q}_{Recuperatore} = \dot{m}_{SCO2} * (h_5 - h_4) = \dot{m}_{SCO2} * (h_7 - h_8)$$

The effectiveness of recuperator is given as:

$$\varepsilon = \frac{(h_5 - h_4)}{(h_7 - h_4)}$$

where h_7 is the enthalpy at the state 7 which is based on the assumption that the temperature of SCO2 stream leaving the recuperator at state 1 reaches the temperature of incoming SCO2 stream from the compressor at state 1.
Now waste heat provided by SCO2 cycle to ORC through the HX unit (process 2 to 3) can be defined as:

$$Q_{heat\ exchanger\ 1} = \dot{m}_{SCO2} * (h_3 - h_2) = \dot{m}_{ORC} * (h_{12} - h_{15})$$

Where $\dot{m}_{ORC}$ is the mass flow rate of working fluid in bottoming ORC unit. Process (8 to 1) for cooler unit can be written as:

$$\dot{Q}_{cooler} = \dot{m}_{SCO2} * (h_8 - h_1)$$

Process 1 to 2 for compressor 1 is given by:

$$W_{compressor\ 1} = \frac{\dot{m}_{SCO2} * (h_{2s} - h_1)}{\eta_{compressor\ 1}}$$

Now thermal process (12 to 13) for ORC turbine is define by

$$W_{Organic\ turbine} = \dot{m}_{ORC} * (h_{12} - h_{13s}) * \eta_{ORC\ turbine}$$

Where h_{13s} is the isentropic enthalpy at of ORC turbine.
Process 13 to 14 for condenser unit can be written as:

$$\dot{Q}_{condensore} = \dot{m}_{ORC} * (h_{14} - h_{13})$$

Process 14 to 15 for pump is given as:

$$W_{ORC\ pump} = \frac{\dot{m}_{ORC} * (h_{15s} - h_{14})}{\eta_{ORC\ pump}}$$

Where h_{15s} is the isentropic enthalpy at the outlet pump.
Further the First law efficiency of the overall cycle can be expressed as:

$$\eta_{first\ law} = \frac{W_{net\ SCO2} + W_{net\ ORC}}{\dot{Q}_{inl}}$$
$$W_{net\ SCO2} = W_{gas\ turbine} - W_{compressore\ 1} - W_{compressore2}$$
$$W_{net\ ORC} = W_{Organic\ turbine} - W_{Organic\ pump}$$

So net work output for overall plant is given by:

$$W_{net\ overall\ plant} = W_{net\ SCO2} + W_{net\ ORC}$$

Now outlet temperature of SPTC can be assumed as constant. Therefore, total exergy input to the cycle is given by:

$$Ex_{inl} = Q_{inl} * \left[1 - \frac{T_0}{T_{11}}\right]$$

Apart from these, few important terms like exergy destruction ratio, net work output can also be calculated. Exergy destruction ratio is also an important exergetic parameter for the improvement in system performance. It is defined as the ratio of rate of exergy destruction to the inlet exergy. Overall plant exergy destruction ratio is given as:

Exergy destruction ratio (EDR) = $\frac{Ed_{with\ SPTC}}{Ex_{inl}}$

4. Results and Discussion

In this study, a comprehensive exergetic analysis for individual component of the considered system is presented first. Then the exergetic performance of SPTC driven combined cycle (ORC and SCO2) has been examined against the variations of solar irradiation intensity and inlet pressure of ORC turbine by using EES software. Currently, SPTC system is designed on the basis of the average value of direct normal irradiance, i.e. 850 W/m^2 according to Indian climate conditions in which combined (SCO2-ORC) cycle is assumed to be operated. Furthermore, the effect of solar irradiation on the combined cycle performance was examined during the daytime between the full ranges of direct normal irradiance, i.e. 500 W/m^2 to 950W/m^2.

4.1 Effect on system performance with variation in intensity of solar irradiation

The exergetic performance of considered system is clearly affected by the changes in solar irradiation intensity as illustrated in Fig. 4. It has been observed that the exergy efficiency of overall cycle (SPTC-SCO2-ORC) increases with the increase in solar irradiation intensity which has been analyzed under the simulation conditions of high pressure and mass flow rate of SCO2 (i.e. 25 MPa and 2 kg/s). As can be seen, the increasing solar irradiation intensity upon the collector field gives the better and efficient utilization of specific range of the solar collector rows available in the overall solar field, which results in exergetic performance enhances. Among all the selected refrigerants for the overall cycle, benzene showed the maximum exergetic efficiency followed by R236fa, cyclohexane, R245fa, R227ea, R1234yf, R134a, isopentane, R410a and isobutene. Fig.4 indicates that the exergy efficiency of benzene based combined cycle increases continuously from 14.83% at 500 W/m^2 to. 18.01% at 950 W/m^2. While the study revealed that benzene and R236fa have very less difference in second law efficiency18.01% and 18.0% respectively at 500 W/m^2 respectively.

From Fig 5 it can be seen that 'benzene' has the highest overall plant first law efficiency value among the other considered working fluids, which is increases from 10.01% at 500 W/m^2 to 12.09% at 950 W/ m^2. Alternatively, R410a has the lowest value of thermal efficiency which is around 7.21% at 500 W/m^2 increases to 8.75 at 950 W/ m^2. From fig 6 it can be seen that overall plant exergy destruction ratio decreases with increases in solar irradiation as can be seen, the increasing solar irradiation intensity upon the collector field gives the better and efficient utilization of specific range of the solar collector rows available in the overall solar field, which results in exergetic performance enhances consequently decrease in irreversibility so exergy destruction ratio decreases. Exergy destruction ratio is maximum for R410a refrigerant among selected ten refrigerants which is maximum 0.881 at 500 W/m^2 and minimum 0.833 at 950 W/m^2. On other hand exergy destruction ratio is minimum for benzene, 0.8496 at 500 W/m^2 and 0.8169 at 950 W/m$^{2.}$

From fig.7 shows the variation of overall net work output with solar irradiation. Increase in solar irradiation net overall work output also increase. Maximum net work for benzene refrigerant among the other selected refrigerant, it varies from 463.5(KW) at 500 W/m^2 to 1118 (KW) at 950 W/m^2. Minimum net work for R410a refrigerant among other selected working fluids. It varies from 313.3 (KW) at 500 W/m^2 to 863(KW) at 950 W/m^2. But for other refrigerant it varies in between these two refrigerants.

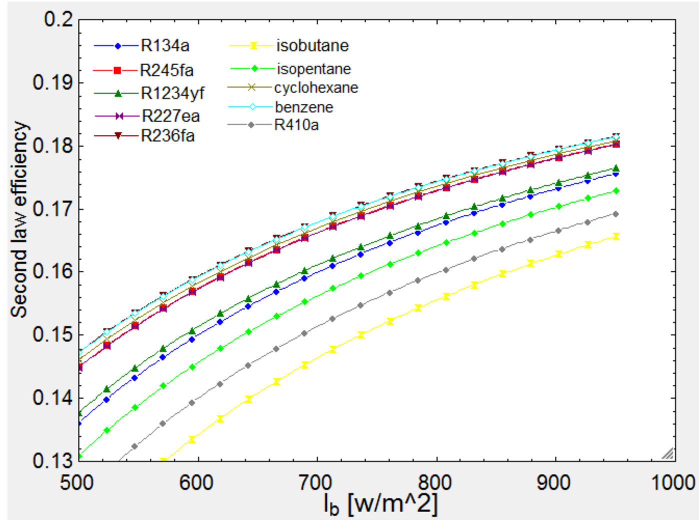

Figure 4: Variation of overall second law efficiency with beam irradiation

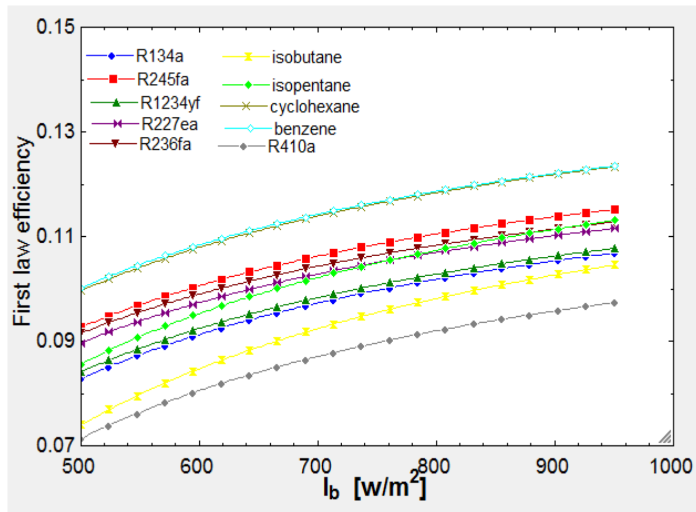

Figure 5: Variation of overall first law efficiency with solar irradiation

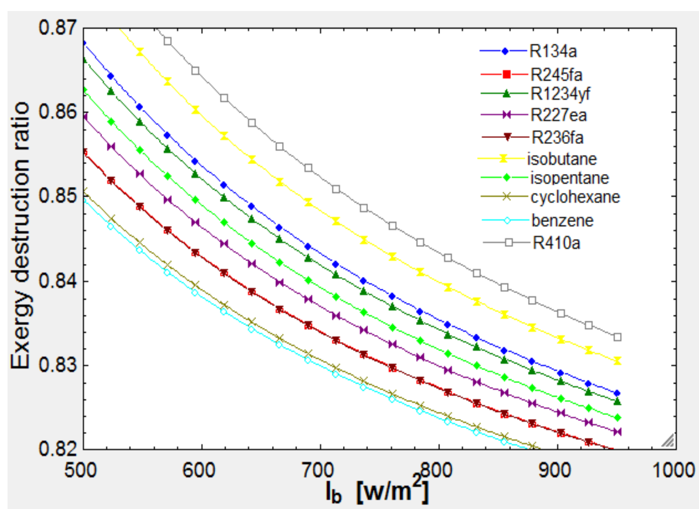

Figure 6: Variation of overall exergy destruction ratio with beam irradiation

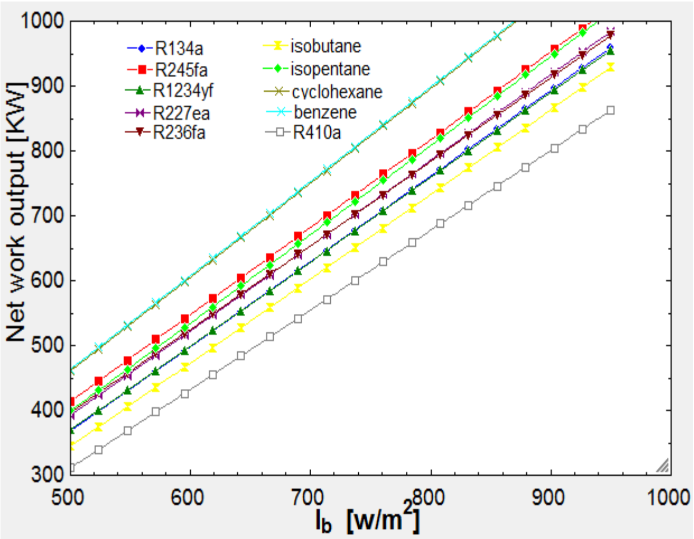

Figure 7: Variation of overall net work output with beam irradiation

4.2 Effect on system performance with variation in HTF velocity in absorber tube

Fig.8 shows the variation of overall second law efficiency with HTF velocity in absorber tube. From figure it can be seen that second law efficiency increases with velocity. Reason for increase in second law efficiency with the velocity is that due to increases in velocity of fluid Reynolds number is increased consequently convective heat transfer coefficient increased so much heat is carried with heat transfer fluids so much heat available with HTF. This leads to increase in second law efficiency. Maximum for R236fa and varies 11% at 0.01(m/s) to 18.35% at 0.1(m/s) and minimum for isobutene among other selected working fluids, it varies from 9.81% at 0.01(m/s) to 17.16% at 0.1(m/s). But for other fluids it varies in between.

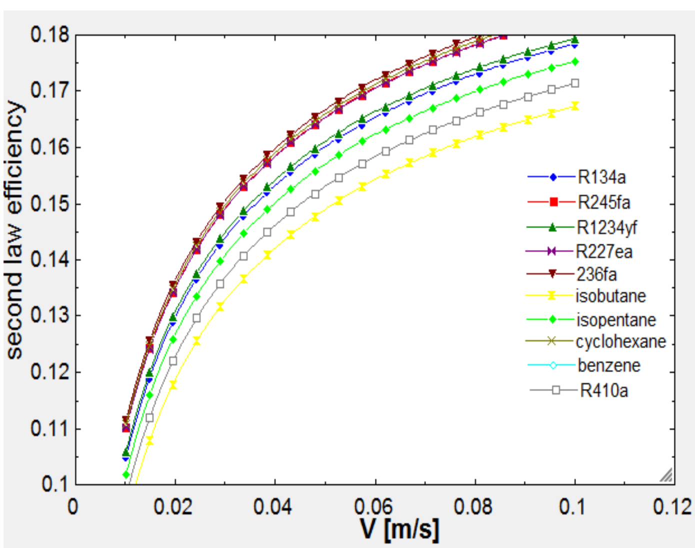

Figure 8: Variation of overall second law efficiency with HTF velocity in absorber tube

Fig.9. shows the variation of overall first law efficiency with HTF velocity. From figure it can be seen that first law efficiency increases with velocity. Reason for increase in first law efficiency with the velocity is that due to increases in velocity of fluid Reynolds number is increased consequently convective heat transfer coefficient increased, so much heat is carried with heat transfer fluids consequently much heat available with HTF. This leads to increase in first law efficiency. Maximum for benzene and varies 11.33% at 0.01(m/s) to 18.68% at 0.1(m/s) and minimum for R410a among other selected working fluids, it varies from 9.48% at 0.01(m/s) to 16.84% at 0.1(m/s). But for other fluids it varies in between.

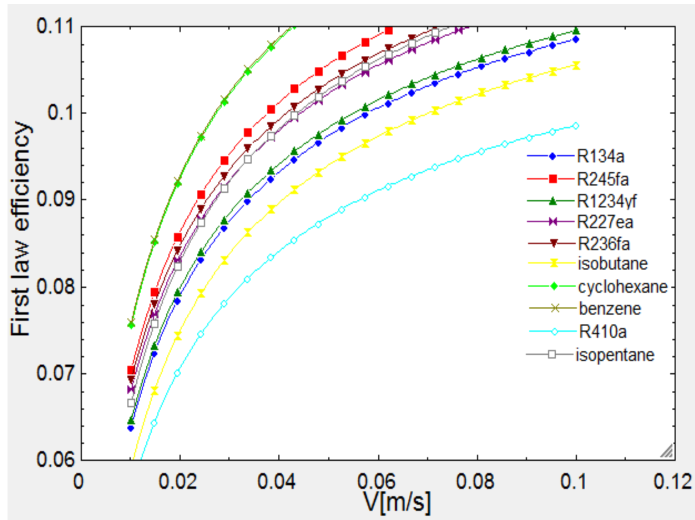

Figure 9: Variation of overall first law efficiency with HTF velocity in absorber tube

Fig.10. Shows the variation of exergy destruction ratio with the HTF velocity. From fig it can be seen that EDR decreases with velocity and maximum for the R410a varies from 0.912 at 0.01 (m/s) to 0.835 at 0.1(m/s). minimum for benzene, varies from 0.886 at 0.01(m/s) to 0.81 at 0.10 (m/s). From fig. 11 it can be seen that the net overall work output increases with velocity of HTF. Maximum for benzene and minimum for R410a among other selected working fluids. For benzene it varies from 420.2(KW) at 0.01 (m/s) to 1020.31(KW) at 0.1(m/s). For R410a it varies from 300.01(KW) at 0.01 (m/s) to 802.32(KW) at 0.1(m/s).

4.3 Effect on system performance with variation of organic turbine inlet pressure

Fig.12 shows the variation of the overall second law efficiency with organic turbine inlet pressure. From the fig it can be seen that efficiency increases slightly with inlet pressure of the organic turbine. Benzene shows maximum efficiency among other selected working fluids. On other hand the R410a shows the minimum efficiency. Also R236fa shows equal efficiency with benzene after approximately 3200(KPa). Fig 13 illustrates the variation of first law efficiency with the inlet organic turbine pressure. It can be seen from figure that benzene shows maximum efficiency and slightly increases with increase in pressure. On other hand R410a shows the minimum efficiency and increases with increase in pressure. Fig14. Shows the variation of exergy destruction with inlet pressure to ORC. From figure it can be seen that benzene shows minimum exergy destruction among the other selected working fluids. But on other hand isobutene shows the maximum exergy destruction. For benzene it varies from the 0.824 at 2500KPa to 0.823 at 4500 Kpa. Fig 15 illustrates the variation of the net work output with ORC inlet pressure. From fig it can be seen that net work output increases with increases in pressure .With increase in pressure expansion ration increases consequently increases in net work output. Benzene and R410a shows that maximum and minimum work output respectively among the other selected working fluids. For benzene it varies from 975 (KW) at 2500 KPa to 976.12(KW). Variation for benzene is very small almost constant but For R410a variation is considerable amount. For R410a it varies from 712.5 (KW) at 2500 KPa to 748 (KW) at 4500 (KPa).

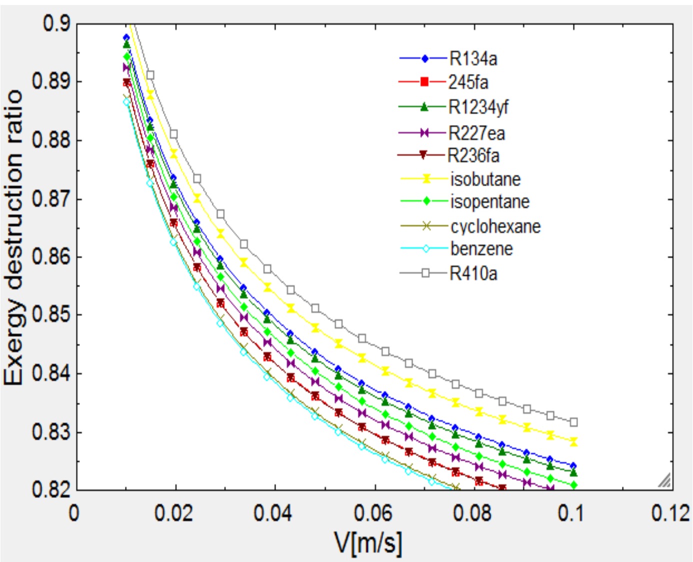

Figure 10: Variation of exergy destruction ratio with HTF velocity in absorber pipe

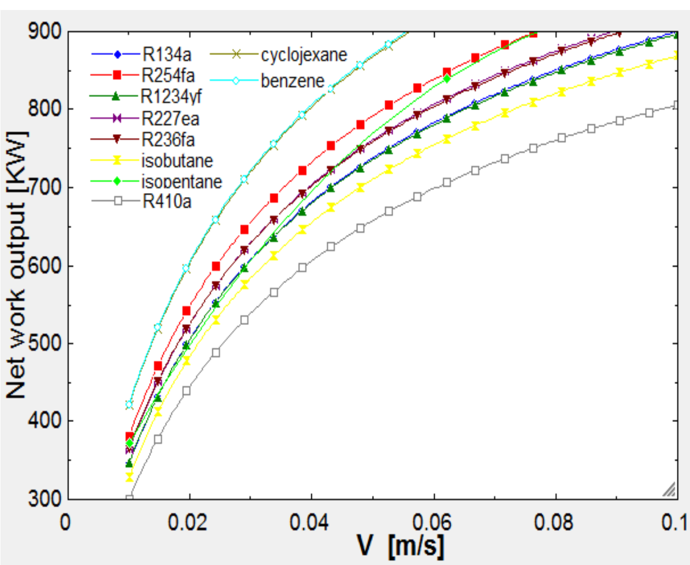

Figure 11: Variation of overall net work output with velocity of HTF in absorber tube

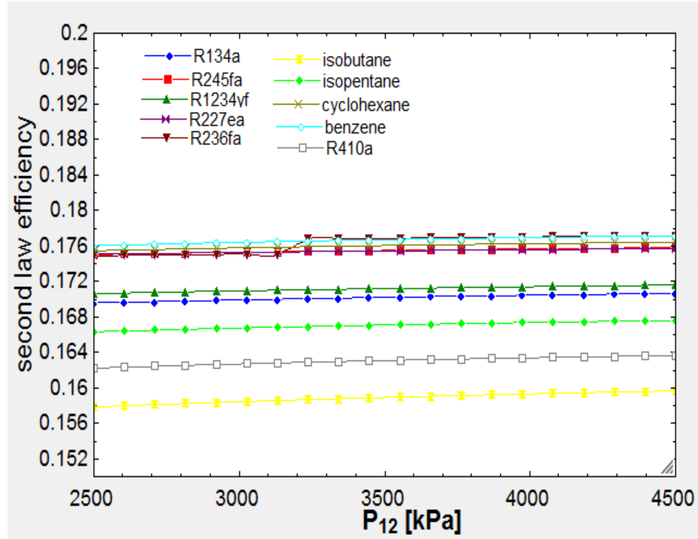

Figure 12: Variation of overall second law efficiency with ORC inlet pressure

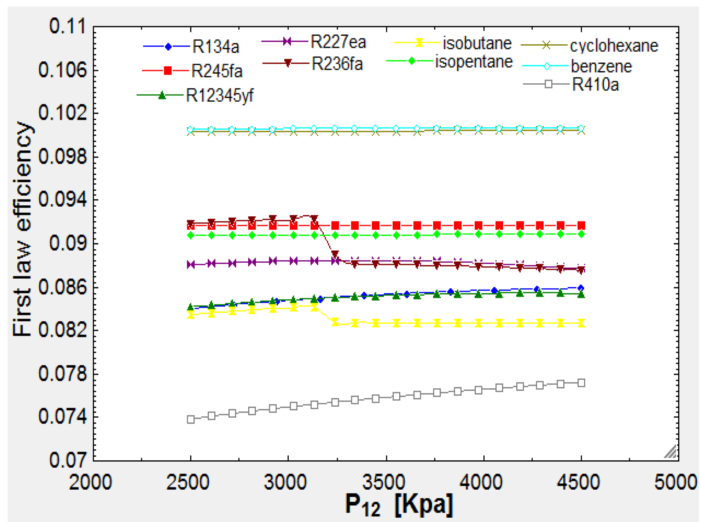

Figure 13: Variation of overall first law efficiency with inlet pressure to ORC

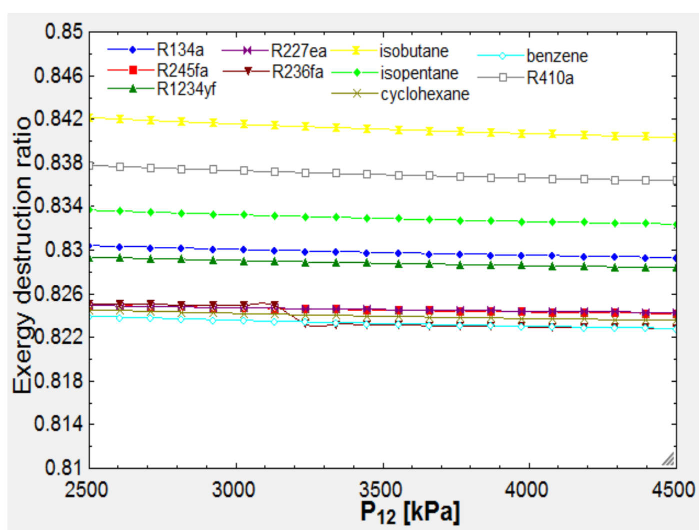

Figure 14: Variation exergy destruction ratio with ORC inlet pressure

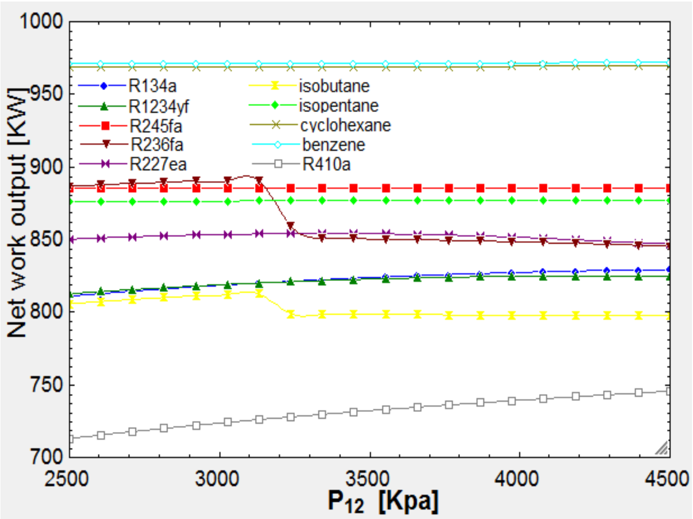

Figure 15: Variation of overall net work output with ORC inlet pressure

5. Conclusions

In this study, complete exergetic (second law) and energetic (first law) analysis of the SPTC integrated combined cycle has been conducted. This research has considered ten various refrigerants for the ORC (i.e. bottoming cycle or intercooler cycle): R134a, R245fa, R1234yf, R227ea, R236fa, isobutene, cyclohexane, isopentane, benzene and R410a. The following conclusions are made

- Exergetic efficiency of the overall cycle (SPTC-SCO2-ORC) increases as the solar irradiation intensity increases. Results of the study conclude that benzene has highest value of second law efficiency for overall cycle which is around 18.01% at 950 W/m^2. While the study revealed that benzene and R236fa have very less difference in second law efficiency18.01% and 18.0% respectively at 500 W/m^2 respectively followed by the R236fa, cyclohexane, R1234yf, and R245fa,R227ea, R134a, isopentane and Isobutene. Alternatively, for isobutene overall cycle has the lowest exergy efficiency value, i.e. 16.87% at 950 W/m^2 which can be due to the maximum amount exergy loss present in this cycle. On the other hand, rate of exergy destruction continuously shows a decreasing trend with the increase in solar irradiation intensity. Thus, benzene has the lowest exergy destruction rate and isobutene has the highest exergy destruction rate.
- In addition, (thermal) first law efficiency of the overall cycle increases as the solar irradiation intensity increases. The study reveals that benzene has the highest thermal efficiency of 12.09% at 950 W/ m^2, while the R410a has the lowest thermal efficiency value of 8.75% at 950 W/ m^2.
- It has also been observed that second law as well as first law performance increases slightly as the inlet pressure of the ORC turbine increases. Therefore, benzene overall cycle gives the best second law and first efficiency which is around 17.61% and 10.1% at 2.5 MPa, respectively. Alternatively, benzene possesses the lowest rate of total exergy destruction ratio as compared to other selected fluids, i.e. 0.824 at 2.5 MPa.
- It has been also observed that first law and second law efficiencies increases with increases in velocity of HTF in solar field.Net work output also increases with increase in velocity of HTF fluid in solar field.
- It has been found that SPTC is the primary source of exergy destruction in which more than 79.11% of the solar inlet exergy has been destructed in the solar collector field only, which is a crucial amount. Therefore, it requires a necessary care during the designing of SPTC plant to decrease down the exergy destruction rate in the solar driven combined cycle.

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