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Thermal performance of combined cycle power plant with solar reheating and regeneration using ecofriendly organic fluids

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Abstract

The use of solar energy in natural gas combined cycle has been demonstrated in various countries since several years. There are many parameters which governed the cost on combined solar integrated cycles that are initially the reintegration of available running power plants. The main objective of the solar integrated power plant is to mitigate the emission and risk associated with the already running power plants .The other important purpose of integration of solar energy system into already running power cycles is to minimize the running cost in the existing equipment.
In this paper, thermodynamic (exergy and energy) analysis of Gas turbine-Organic Rankine combined cycle with the solar reheating of organic fluid is done and the results are compared with combined cycle with the solar reheating, with the simple combined cycle and combined cycle with regeneration of organic fluids. The thermal performance of system is compared with different organic fluids like R134a, R227ea, R245fa, R1234ze and R1234yf at different organic Rankine cycle maximum temperature and maximum pressure. It was observed that R1234yf shows maximum increase in second law efficiency by regeneration around 49.2%. 42.94% using ORC and 48.2%, using ORC with Regeneration and solar Reheat system while R134a shows maximum organic Rankine cycle efficiency of 48.3%. exergetic efficiency of combined cycle with regeneration and reheating using R1234ze organic fluid. However maximum first law efficiency comes out by using R134a fluid. For practical applications and various problems associated with ecofriendly organic fluids (such as flammability and explosion risk should be considered), R134a can be used as better option in regeneration combined cycle plant with solar reheating.

Key words: Organic Rankine Cycles, Regeneration, Solar heating combined cycle power plants, Energy-Exergy Analysis

1. Introduction

Several solar integrated combined cycles (ISCCs) are being used in all the world and there are many projects are processing. The integrated solar combined cycles (ISCCs) have several advantages as compared to solar thermal power plants, because these give higher conversion solar efficiency, and it have very low investment cost. Many entrepreneurs and owners are ready to invest the many due to it low risk associated with the smaller plants as compared to the solar thermal power plants.
The procedure for changing over the vitality in a fuel into electric power incorporates the formation of mechanical work, which is then changed over into electric power by a generator. Contingent upon fuel sort and thermodynamic process, the in general productivity of this change can be as low as 33 %. This implies 67% of the inert vitality of the fuel goes up squandered. For instance, steam electric power plants which utilizes boilers to combust a petroleum derivative normal 33% efficiency. Basic cycle gas turbine (GTs) plants normal just shy of 30 percent productivity on gaseous petrol, and around 25 % on fuel oil. Quite a bit of this squandered vitality misfortunes as warm vitality in the hot fumes gasses from the ignition procedure. To build the general proficiency of electric power plants, various procedures can be consolidated to recoup and use the leftover warmth vitality in hot fumes gasses. In joined cycle mode, control plants can accomplish electrical efficiencies up to 60 %. The expression "joined cycle" alludes to the consolidating of numerous thermodynamic cycles to create control. Joined cycle operation utilizes a heat recovery boiler (HRB) that assimilates warm from high temperature deplete gasses to deliver natural vapor, which is then provided to a natural turbine to produce additional electric power. The procedure for making vapor to create work utilizing a natural

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turbine depends on the Rankine cycle. The most widely recognized sort of combined cycle control plant uses gas turbines and is known as a combined cycle gas turbine (CCGT) plant. Since gas turbines have low effectiveness in straightforward cycle operation, the yield delivered by the steam turbine represents about portion of the CCGT plant yield. There are various designs for CCGT control plants, yet normally each GT has its own related HRB, and different HRBs supply vapor to at least one organic turbines. Energy-exergy analysis gives a uniform base for correlation of different thermodynamic procedures. This analysis demonstrates the data with respect to losses that incorporate their area subjectively and quantitatively. This data can be utilized for further change in the outline and operation of the framework. By finding the exergy destruction in the each component and exergy destruction ratio of whole system, the system thermodynamic performance can be enhanced by enhancing the exegetic performance of the component and the whole system. The scope and purpose of this paper is to develop effective methodology to achieve energy-exegetic optimizations of CCGT power plants to improve thermodynamic performance of the power plant by integration of ORC and energy-exergy optimization method. With the help of this method, it would be possible to (a) provide information about the exergy destruction and exergy losses along with their location. b) Predict the highest exergy destruction components of the system. c) Suggest ways of improving the exergetic efficiency. d) Find the optimal values of operating parameters, which gives the maximum possible power output and thermal efficiency so that it would be possible to calculate minimum possible exergy destructions in the component by modifications in existing design.

2. Literature Review

In order to have an idea of the present methodology development in the area of performance and optimization of combined cycle gas turbine power plant, a brief survey of available literature was made. However, this chapter is concerned with a review of literature on optimization performed on various thermal systems. In general, some authors focus on the gas turbine operating parameters (topping cycle), others optimize the steam plant (bottoming cycle) based on a given gas turbine, whereas others propose appropriate optimization methods for the whole combined cycle power plant without integration of ORC. Furthermore, the optimization can be analyzed from a thermodynamic point of view, according to the first and/or second law analysis, or using a thermo economic or environmental-economic strategy Jaafar et al [1], Dincer et.al.[2],Boyano et al [3] , Cabrera M., et al [4] etc. From the point of view of optimization methodology, there are many types of analyses. In this work, the review will highlight most common methodology: the exergy destruction method, and the efficiency improvement method.

2.1 Thermodynamic Analysis and Optimization

The gas turbine operating parameters which influence the combined cycle gas turbine performance are; ambient conditions, compressor pressure ratio, and turbine inlet temperature. One of the factors that affect gas turbine performance is the ambient conditions, mainly ambient temperature, atmospheric pressure, and the relative humidity of air. These parameters affect the generated electric power and the heat-rate during operation. The location of power plant plays a major role on its performance. The atmospheric air, which enters the compressor, becomes hotter after compression and it is directed to a combustion chamber. Several authors reported the effect of ambient temperature: Ibrahim et al [5], Ameri and Hejazi [6],Boonnasa et al [7] and Hosseini et al [8]. The properties of air entering combustion chamber depend upon the compressor pressure ratio studied. Ibrahim and Rahman [9], and Khaliq and Kaushik et al [10] performed a parametric thermodynamic analysis of a combined cycle gas turbine. They investigated the effect of operating parameters, compression ratio, gas-turbine peak temperature ratio, isentropic compressor and efficiency and air fuel ratio on the overall plant performance. Their results show that the compression ratios, air to fuel ratio as well as the isentropic efficiencies are strongly influenced by the overall thermal efficiency of the combined cycle gas turbine power plant. The overall thermal efficiency increases with compression ratio as well as isentropic compressor and turbine efficiency. However, the variation of overall thermal efficiency is minor at the lower compression ratio while it is very significant at the higher compression ratio for both isentropic compressor and turbine efficiency. The overall efficiencies for combined cycle gas turbine are much higher than the efficiencies of gas turbine plants power output decreases linearly with the increase temperature. The efficiency of steam power plants can be improved by increasing the live steam and reheat-steam parameters, and by introducing high-efficiency, low-loss turbine blade geometries. The first goal, to increase the steam parameters, is primarily achieved by choosing appropriate materials for the components operating under live-steam and reheat-steam conditions while retaining the proven designs. Collaborative European programs have led to the development and qualification of steels with much improved creep properties at temperatures of up to 600 °C, appropriate for the manufacture of key components. At the same time, optimization of the blade profiles and geometries allowed further major improvements in operating efficiency. The achievable improvements in efficiency is about 0.5% per 10 °C live steam and reheat (RH) temperature increase, and 0.2 % per 10 bar pressure increase. Second important part of the bottoming cycle is the heat recovery steam turbine (HRSG), its design and optimization affects to a large extent influence the efficiency and the cost of the whole plant.

Mohagheghi M [11] carried out thermodynamic optimization of design variables and properties of air entering combustion chamber depend upon the compressor pressure ratio heat exchangers layout in a heat recovery steam generator HRSG for combined cycle gas turbine CCGT using a genetic algorithm. Their method was introduced for modelling the steam cycle in advanced combined cycles by organizing the non-linear equations and their simultaneous used solutions with numerical methods. 8 In addition to the optimization of design variables of the recovery boiler, they performed the distribution of heat exchangers among different sections and optimized their layouts in HRSGs. A standard gas turbine was assumed, and then outlet gas stream conditions (mass flow rate, temperature, and chemical composition of gas stream) were considered as the inlet parameters for the recovery boiler model. From the optimization process maximum output power from a steam cycle for different HRSGs was then analyzed. Bracco S [12] studied the combined cycle power plant with a single level heat recovery steam generator HRSG. They developed a mathematical model to determine the optimal steam pressure values in the HRSG according to different objective functions (in the HRSG for a given gas turbine). Their work reports numerical results for the combined cycle power plant considering four different gas turbines. The optimization approach was focused on the study of the heat transfer between the steam and the exhaust gas in the HRSG, based on an exergetic analysis and comparison among different objective functions that refer to the HRSG specifically or to the whole bottoming cycle. In their mathematical model, they considered the presence of specific constraints for the operating parameters of the power plant, the most important constraints that were considered refer to the steam quality at the turbine outlet, the HRSG outlet exhaust gas temperature and the steam turbine blade height. In their work, a parametric analysis was also performed to evaluate the influence of the gas temperature at the HRSG inlet and the pinch point temperature difference on the considered objective functions. Woudstra et al [15**]** performed the thermodynamic evaluation of combined cycle plants with the same gas turbine and different steam bottoming cycles. The evaluation showed that the increasing the number of pressure levels of steam generation will reduce the losses due to heat transfer in the HRSG, but also the exergy loss due to the exhaust of flue gas to the stack. Among the investigated configurations for bottoming cycle, triple pressure reheat was the best option from exergy point of view. Mansouri et al [16] investigated the effect of pressure levels of steam generation at heat recovery steam generator HRSG on the energetic and exergetic efficiency of HRSG, bottoming cycle and combined cycle power plants, as well as the effect of Xiang and Chen [17]considered a combined cycle with three-pressure HRSG, equipped with the GE PG9351FA gas turbine. They maximized the combined cycle efficiency through the optimization of the HRSG operating parameters by minimizing exergy losses. Moreover, they highlighted the influence of the HRSG inlet gas temperature on the bottoming cycle efficiency. They studied the influence of HRSG inlet gas temperature on the steam bottoming cycle efficiency. Their result shows that increasing the HRSG inlet temperature has less improvement to steam cycle efficiency when it is over 590ºC. Kelly et al [18] demonstrated that the most efficient way for converting solar thermal energy into electricity is to withdraw feed water from the heat recovery steam generator (HRSG) downstream of the last economizer, to produce high pressure saturated steam and to return the steam to the HRSG for superheating and reheating. The integrated solar plant concept offers an effective means for the continued development of parabolic trough technology. In a careful plant design, solar thermal to electric conversion efficiencies will exceed, often by a significant amount, those of a solar-only parabolic trough project. An integrated plant bears only the incremental capital cost of a larger Rankine cycle which provides further reductions in the leveled cost of solar energy. He YaLing et al [19] proposed a model for a typical parabolic trough solar thermal power generation system with Organic Rankine Cycle (PT-SEGS–ORC) was built within the transient energy simulation package TRNSYS. They found that the heat loss of the solar collector increases sharply with the increase in Pinter at beginning and then reaches to an approximately constant value. The variation of heat collecting efficiency with v is quite similar to the variation of heat losses.. In addition, it is found that the optimal volume of the thermal storage system is sensitively dependent on the solar radiation intensity. Gang et al.[20]the innovative configuration of low temperature solar thermal electricity generation with regenerative organic Rankine cycle (ORC) mainly consisting of small concentration ratio compound parabolic concentrators (CPC) and the regenerative ORC. The effects of regenerative cycle on the collector, ORC, and overall electricity efficiency are then analyzed. The results indicate that the regenerative cycle has positive effects on the ORC efficiency but negative ones on the collector efficiency due to increment of the average working temperature of the first-stage collectors. And found that there generative cycle optimization of the solar thermal electric generation differs from that of a solo ORC. The system electricity efficiency with regenerative ORC is about 8.6% for irradiance 750 W/m2 and is relatively higher than that without the regenerative cycle by4.9%.Goswami DY [21] proposed co-generation system producing electricity and fresh water by a solar field driven supercritical organic Rankine cycle (SORC) coupled with desalination. The proposed system can use parabolic trough solar collectors (among other options) to produce 700 kW thermal energy with temperatures up to 400°C at peak conditions. Thermal energy is delivered to the SORC which uses hexamethyldisilane (MM) as the working organic fluid and could achieve cycle efficiency close to 21%. The SORC condensation process is undertaken by the feed seawater to reduce thermal pollution. Due to the elevated temperature of the preheated seawater, the RO unit specific energy consumption decreases. Although, lot of literature it is found that the efficiency of the combined cycle is more than the simple individual cycle. Other important conclusion found that the more and more energy going to waste from stack with exhaust flue gases even after passing through HGRC. Almost flue gas around 150-1800C is going to waste from stack.

After reading literature review it is concluded that no researcher use the energy at temperature 150-1800C from the flue gases. In this research integration of the ORC (organic Rankine cycle) in the preexisting cycle is done for recovery of the low temperature heat from the exhaust gases which are coming from the HRSG after generation of the steam for simple Rankine cycle. It is proposed to examine the effect of the various parameters on the performance of the combined cycle and comparison is done with or without integration of the ORC.

2.2 Organic Rankine Cycle

Organic Rankine cycles have gotten more consideration amid the most recent decade. This cycle takes after the crucial principles of regular Rankine cycles working with steam in like manner plants however has a few points of interest over steam Rankine cycle which made it prevalent. Firstly this cycle can work on low pressures and temperatures in comparison to the conventional Rankine cycle and reveals a better result than steam Rankine cycle especially from low grade heat sources because it has working fluids include such as variety of HCs and other refrigerants what's more, as per scope of open heat source pressure and temperatures, different outputs can be obtained by using useful working fluids, secondly, it can also work without multi-stage turbines and feed-water heaters and that thing makes it simple using. Although this, solar parabolic collectors are a tremendous source of heat energy but these have low grade thermal energy. Because of this, these solar collectors give only some KWs to some megawatts of power generation mainly near factories and rural areas to generate own electricity consumption without the necessity for connection to grid that may be costly as shown in Fig-1.

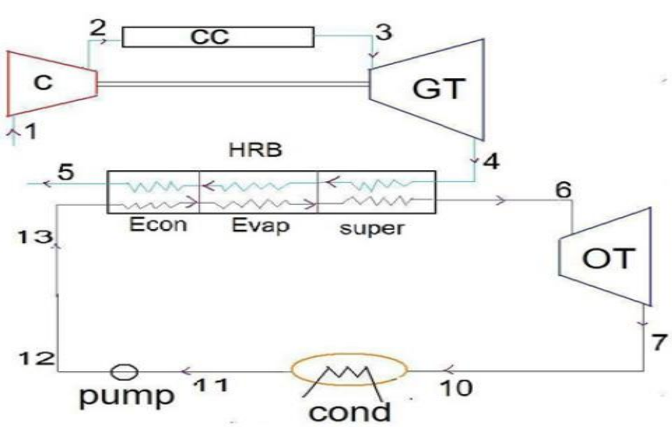

Figure 1: Combined Cycle plant without regeneration

Disadvantages of solar ORCs are comparatively high costs and low thermal efficiency (10 to 25 %) according to working fluids and working conditions) mainly because of low HTF (Heat transfer fluids) temperature in solar collector. As described before, the organic fluid works in ORC cycles are classified into HCs and refrigerants, some of those are dry liquids which mean they have a positive slope T-S graph in the vapour area. This makes it reasonable for some organic liquids to works legitimately without superheating to an extraordinary possibility and make no harm turbine. It has been appeared in this review, an examination of various dry organic liquids with or without superheating and recuperation has been done to reveal the difference in cycle effectiveness and execution of the system that encourages us to settle on a choice to pick the system condition as indicated by our requirements.

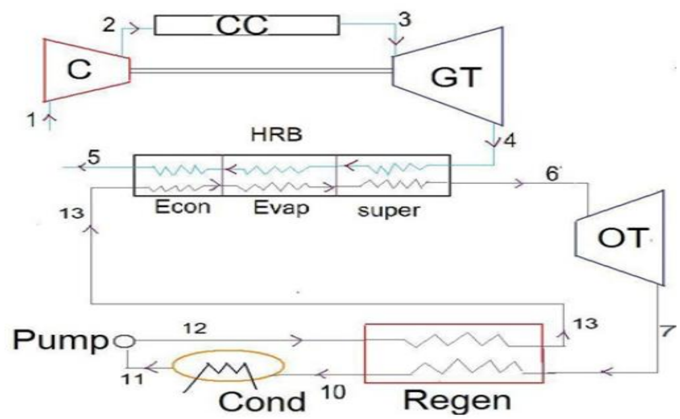

Figure 2: Combined cycle plant with regeneration

2.3 Combined Cycle Principles of Operation

The organic Rankine cycle works on the principle of a turbo generator which works as a simple steam turbine to convert thermal power into mechanical power and then into electrical power by an electrical generator. Besides of using steam of water, orc system evaporates organic fluid, classified by a molecular weight higher than the water that leads to slow rotation of the organic turbine that also leads to the less erosion of metallic parts and the blades. Organic Rankine cycle is normally a Rankine cycle in which besides of using water other organic fluid is used like R134a, R1234ze, R1234a, R245fa etc as shown in Fig-3.

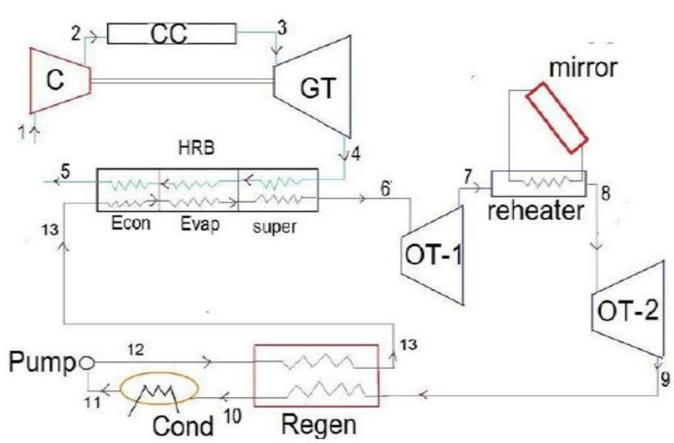

Figure 3: Combined cycle plant with regeneration and solar reheating

The heat recovery boiler (HRB) is nothing but is the simple heat exchanger, or it can be used as a series of heat exchangers. It is also can be called a boiler, because it produces organic vapour for the ORC turbine by passing the hot fumes gas spilling out of a gas turbine or ignition motor through edge of heat exchanger tubes. The HRB can takes a shot at common flow and use constrained dissemination utilizing pumps. As the hot fumes gasses stream past heat exchanger tubes through which boiling water courses, heat is utilized causing the made of vapor in the tubes. The tubes are masterminded in modules or segments, each serving an alternate capacity in the creation of dry superheated vapor. These modules are alluded to as evaporators, super heater and preheaters and economizers. The economizer works as a heat exchanger which preheats the organic fluid (liquid) to get the temperature to the saturation temperature (boiling point), that liquid to be supplied to a thick-walled boiler drum. That drum is installed where finned evaporator tubes is located that circulate heated organic fluid. Hot exhaust gases passing past to the evaporator tubes, and the heat is being absorbed and then the vapor is being created of in the tubes. This vapor-liquid mixture goes to the boiler drum where the vapour is separated from the hot liquid by using moisture cyclones and separators. The separated liquid is again recirculated to the evaporator tube the function of the some boiler drums is also to storage and water treatment functions. There are several other design of steam boiler in which another design is a once-through HRB, where thin-walled components are used in place of boiler drum which are better for handling changes in exhaust gas temperature and vapour pressures during frequently stops and starts. In other designs, duct burners can be used for adding heat to the exhaust gas current and boost vapour manufacturing; it's have been used to generate vapour even if no not sufficient exhaust gas flow is there. Saturated vapour from boiler drums or any once-through boiler system has been sent to super heater to produce dry vapour that is requirement for the organic turbine. And the organic fluids goes to the preheaters, that are placed where very low amount of heat is available and then this fluid takes energy from three and heated by the heat exchanger liquids such as the mixture of glycol water and from there very economic and useful amount of heat is extracted. The superheated organic vapour produced from the HRB is supply to organic turbine where it is expanded by the turbine blades and gives rotation to turbine shaft. The mechanical energy delivered to the generator driving shaft is transformed into high grade energy (electricity). After exiting the organic turbine, the organic vapor goes to the condenser which paths the condensed organic liquid back to HRB Combined-cycle plant works to generate electricity and absorbs waste heat from gas turbine to increase efficiency and the electrical output. Gas turbine plant burns fuel and air is compressed in compressor and then it mixes with the fuel combustion chamber. This mixture of gas and fuel is goes to gas turbine. This hot mixture is rotates the turbine blade and then shaft of turbine is rotated. This rotating turbine gives its mechanical energy to the generator shaft which transformed a part of rotating energy into electrical power. The heat recovery boiler absorbs exhaust gas. This Heat Recovery Boiler (HRB) absorbs heat from the exhaust gas turbine which would otherwise goes through the exhaust stack. The HRB makes organic vapor of organic fluid from the heat of exhaust from the gas turbine and it is supplied to steam turbine. Organic turbine gives additional electricity. The organic turbine sends its mechanical energy to the generator drive shaft, and then the additional electrical power is produced as shown in Fig-4 and 5 respectively.

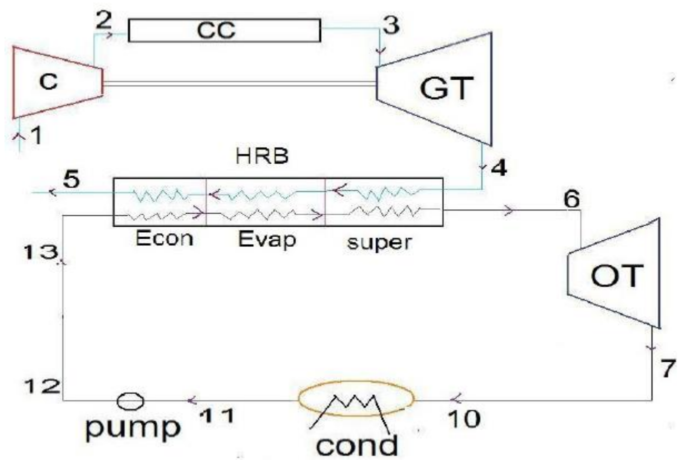

Figure 4: Combined cycle plant without regeneration

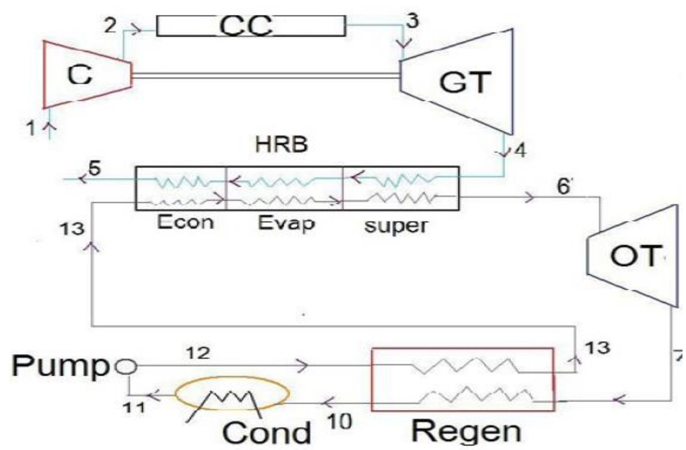

Figure 5: Combined cycle plant with regeneration

2.4 Thermodynamic Analysis

In this paper, a parametric study with various temperature and pressure at organic turbine inlet has been considered to determine efficiency and performance of organic fluid in system. The following assumptions are there to simplify the analysis, also taking energy analysis.

1. Assumed all the components are steady-state process and steady flow.

2. The changes in the kinetic energy and the potential energy are assumed to be negligible.
3. There are negligible heat and pressure loss in pipes that are connecting all the components to each other.
4. All compressor, turbines, and pump work adiabatically.
5. Pressure drops in regenerator, HRB, and condenser neglected.

Considered as control volume.

Mass Balance

$\sum_{in} m_{in} = \sum_{out} m_{out}$

Energy Balance

$Q\text{-}W + \sum_{in} m_{in} - \sum_{out} m_{out} = 0$

2.5 Energy changes in the each component of combined cycle plant

2.5.1 Air Compressor

Compressor is a work absorbing device. Air compressor is used for compressing air from atmospheric condition to high pressure. Isentropic work input to the compressor is expressed as

$W_c = \dot{m}_a * (h_{2s} - h_1)$

Compressor efficiency as,

$\eta_c = \frac{h_{2s}-h_1}{h_2-h_1}$

Actual compressor work is given by

$W_c = \dot{m}_a * (h_2 - h_1)$

2.5.2 Combustion chamber

In combustion chamber fuel is burnt and heat released by combustion is supplied to compressed air in External heat exchanger at constant pressure. Heat supplied in combustion chamber is given by

$Q_{s1} = \dot{m}_a * (h_3 - h_2)$

2.5.3 Gas turbine

Turbine is a work producing component. Air is expanded adiabatically in gas turbine. Isentropic work output of gas turbine is

$W_{GTi} = \dot{m}_a * (h_3 - h_4)$

Gas turbine efficiency as,

$\eta_{GT} = \frac{h_3-h_4}{h_3-h_{4s}}$

Actual gas turbine work is

$W_{GT} = \dot{m}_a * (h_3 - h_4)$

2.5.4 Heat Recovery Boiler

In HRB energy of hot exhaust gas of gas turbine is used for producing superheated vapor of organic fluid. HRB is a heat exchanger in which heat is transferring from hot exhaust gas to organic fluid. Energy balance for HRB is

$\dot{m}_a h_4 + \dot{m}_f h_{13} = \dot{m}_a h_5 + \dot{m}_f h_6$

Heat supplied to organic fluid,

$QHRB = \dot{m}_f * (h_6 - h_{13})$

2.5.5 Organic Rankine Turbine

Organic turbine is a work obtaining device in which organic fluid is expanded from HRB pressure to condenser pressure adiabatically. The isentropic work output of ORT,
$W_{OTi} = m_f * (h_6 - h_{7s})$
Organic turbine efficiency as,

$\eta_{OT} = \frac{h_6-h_7}{h_6-h_{7s}}$

Actual organic turbine work is

$W_{OT} = \dot{m}_f * (h_6 - h_7)$

2.5.6 Condenser

Condenser is a heat exchanger in which heat is rejected to environment is given by
$Q_{cond} = m_f * (h_{10} - h_{11})$

2.5.7 Organic pump

Organic pump is used for increasing pressure of organic fluid from condenser pressure to boiler pressure. Ideal work of organic pump

$W_{OPi} = m_f * v_{11} * (P_{12} - P_{11})$

Organic pump efficiency as,

$\dot{\eta}_{OP} = W_{OPi} / W_{OP}$

Actual organic pump work is given by

$W_{OP} = m_f * v_{11} * (P_{12} – P_{11})/\dot{\eta}_{OP}$

2.5.8 Efficiency of the Gas Turbine cycle

The efficiency of gas turbine plant is given by the ratio of net-work output of gas turbine plant to the heat supplied in combustion chamber,

$\dot{\eta}_{GTP} =(W_T – W_C)/ Q_{S1}$

2.5.9 Efficiency of Organic Rankine cycle

It is ratio of net work output of ORC and the total heat supplied in ORC

$\dot{\eta}_{ORC} = (W_{OT} – W_{OP})/ Q_{S2}$

2.5.10 Efficiency of the combined cycle plant

The ratio of the total work output combined cycle plant and total external heat supplied

$\dot{\eta}_{CCP} =\{ (W_T – W_C) + (W_{OT} – W_{OP}) \}/Q_{S1}$

2.6 Energy changes in each component of combined cycle plant with regeneration

2.6.1 Air Compressor

Compressor is a work absorbing component. Air compressor is used for compressing air from the atmospheric condition to the high pressure. Isentropic work input to compressor is expressed as

$W_C = m_a * (h_{2S} – h_1)$

Compressor efficiency as

$\eta_C=\frac{(h2S - h1)}{(h2 - h1)}$

Compressor work is specified by

$W_C = m_a * (h_2 – h_1)$

2.6.2 Combustion chamber

In the combustion chamber fuel is burnt and heat released by the combustion is supplied to compressed air in the External heat exchanger at constant pressure.
Heat supplied in the combustion chamber is given by

$Q_{S1} = m * (h_3 – h_2)$

2.6.3 Gas turbine

Air is expanded adiabatically in gas turbine. Isentropic work output of gas turbine is given by

$W_{GTi} = m_a * (h_3 – h4)$

Gas turbine efficiency as,

$\dot{\eta}_{GT} =(h3 – h_4)/(h_3 – h_{4S})$

Actual gas turbine work is

$W_{GT} = m_a * (h_3 – h_4)$

2.6.4 Heat Recovery Boiler

In HRB heat of the hot exhaust gas of gas turbine is used for producing superheated vapor of organic fluid. HRB is a heat exchanger in which heat is transferring from hot exhaust gas to organic fluid. Energy balance for HRB is

$m_a h_4 + m_f h_{13} = m_a h_5 + m_f h_6$

Heat supplied to organic fluid,

$Q_{HRB} = m_f * (h_6 – h_{13})$

2.6.5 Organic Rankine Turbine

Organic turbine is the work producing component in which expansion of organic fluid takes place from boiler pressure to condenser pressure adiabatically. Isentropic work output of ORT,

$W_{OTi} = m_f * (h_6 – h_{7s})$

Organic turbine efficiency as,

$\dot{\eta} = (h_6 – h_7) /(h_6 – h_{7S})$

Actual organic turbine work is

$W_{OT} = m_f * (h_6 – h_7)$

2.6.6 Regenerator

Regenerator is a liquid-vapor heat exchanger in which feed organic liquid is heated by the superheated vapor leaving from organic turbine.

$\dot{m}_f h_9 + \dot{m}_f h_{12} = \dot{m}_f h_{10} + \dot{m}_f h_{13}$

2.6.7 Condenser

Condenser is a heat exchanger in that heat is rejected to environment is given by

$Q_{cond} = m_f * (h_{10} - h_{11})$

2.6.8 Organic pump

Organic pump is used for increasing the pressure of organic fluid from condenser pressure to boiler pressure. Ideal work of organic pump

$W_{OPi} = m_f * v_{11} * (P_{11} – P_{12})$

Organic pump efficiency as,

$\dot{\eta}_{OP} = W_{OPi} / W_{OP}$

Actual organic pump work is given by

$W_{OP} = \{m_f * v_{11} * (P_{12} – P_{11})\}/\dot{\eta}_{OP}$

2.6.9 Efficiency of the Gas Turbine cycle

The efficiency of the gas turbine plant is given by the ratio of the net work output of gas the turbine plant and heat supplied in combustion chamber,

$\dot{\eta}_{GT} = (W_T – Wc)/ Q_{S1}$

2.6.10 Efficiency of Organic Rankine cycle

It is the ratio of the net work output of ORC and total heat supplied in ORC

$\dot{\eta}_{ORC} = (W_{OT} – W_{OP})/ Q_{S2}$

2.6.11 Efficiency of combined cycle plant

It is defined as the ratio of total work output of the combined cycle plant to and total external heat supplied

$\dot{\eta}_{CCP} =\{ (W_T - W_C) + (W_{OT} - W_{OC}) \}/Q_{S1}$

2.7 Energy changes in the each component of combined cycle plant with regeneration and solar reheating

2.8 Air Compressor

Compressor is the work absorbing device. Air compressor is used for compressing the air from atmospheric condition to the high pressure. Isentropic work input to the compressor is expressed as

$W_c = m_a * (h_{2S} – h_1)$

Compressor efficiency as

$\dot{\eta}_c = (h_{2S} – h_1)/ (h_2 – h_1)$

Actual compressor work is specified by

$Wc = m_a * (h_2 – h_1)$

2.8.1 Combustion chamber

In combustion chamber fuel is burnt and the heat released by the combustion is supplied to compressed air in External heat exchanger at constant pressure .Heat supplied through combustion chamber is given by

$Q_{S1} = m_a * (h_3 – h_2)$

Gas turbine: turbine is a work producing device. Air is expanded adiabatically in gas turbine. Isentropic work output of gas turbine is

$W_{GTi} = m_a * (h_3 – h4)$

We have gas turbine efficiency as,
Actual gas turbine work is

$\dot{\eta}_{GT} =(h3 – h_4)/(h_3 – h_{4S})$

2.8.2 Heat Recovery Boiler

In the HRB heat of the hot exhaust gas of the gas turbine is used for producing the superheated vapor of organic fluid. HRB is a heat exchanger in which heat is transferring from the hot exhaust gas to organic fluid. Energy balance for HRB is

$m_a h_4 + m_f h_{13} = m_a h_5 + m_f h_6$

Heat supplied to organic fluid,

$Q_{HRB} = m_f * (h_6 – h_{13})$

2.8.3 Organic Rankine Turbine1

Turbine is a work producing component.the expansion is taking place from HRB pressure to intermediate pressure in organic turbin-1 adiabatically. The isentropic work output,

$W_{OTi} = m_f * (h_6 – h_{7s})$

Organic turbine efficiency as,

$\dot{\eta} = (h_6 – h_7) /(h_6 – h_{7S})$

Actual organic turbine work is

$W_{OT} = m_f * (h_6 - h_7)$

2.8.4 Solar reheater

Solar reheater is the concentrated solar plate collector in that solar energy is used for reheating organic fluid. Heat supplied to the solar collector is given by

$Q_{S2} = A_S I_S \dot{\eta}_m$

Heat supplied to organic fluid

$Q_{S2} = m_f * (h_7 - h_8)$

2.8.5 Organic Rankine Turbine2

Turbine is a work producing component. The expansion of organic fluid is takes place from boiler pressure to intermediate pressure in organic turbin-2 adiabatically. The isentropic work output,

$W_{OT2} = m_f * (h_8 - h_{9S})$

Organic turbine efficiency as,

$\dot{\eta}_{OT2} = (h_8 - h_9)/(h_8 - h_{9S})$

Actual organic turbine work is

$W_{OT} = m_f * (h_8 - h_9)$

2.8.6 Regenerator

Regenerator is a liquid-vapour heat exchanger in which feed organic liquid is heated by superheated vapour leaving from organic turbine.

$m_f h_9 + m_f h_{12} = mfh_{10} + m_f h_{13}$

2.8.7 Condenser

Condenser is a heat exchanger in that heat is rejected to environment is given by

$Qcond = mf * (h_{10} - h_{11})$

2.8.8 Organic pump

Pressure of organic fluid is increased from condenser pressure to boiler pressure by pump. Ideal work of organic pump

$W_{OPi} = m_f * v_{11} * (P_{12} - P_{11})$

Organic pump efficiency as,

$\dot{\eta}_{op} = Wopi/Wop$

Actual organic pump work is given by

$W_{OPi} = \{m_f * v_{11} * (P_{12} - P_{11})\}\dot{\eta}_{OP}$

2.8.9 Efficiency of the Gas Turbine cycle

The efficiency of the gas turbine plant is given by the ratio of network output of the gas turbine plant and heat supplied in combustion chamber,

$\dot{\eta}_{GT} = (W_T - Wc)/Q_{S1}$

2.8.10 Efficiency of Organic Rankine cycle

It is ratio of the net-work output of ORC and total heat supplied in ORC

$\dot{\eta}_{ORC} = (W_{OT1} + W_{OT2} - WC)/(Q_{S2} + Q_{HRB})$

2.8.11 Efficiency of combined cycle plant

The ratio of the total work output of the combined cycle plant to the total heat externally supplied

$\dot{\eta}_{CCP} = [(W_{GT} - W_C) + (W_{OT1} + W_{OT2} - W_{OP})]/(Q_{S1} + Q_{S2})$

3. Exergy analysis of the each component of the combined cycle plant with the regeneration and the solar reheating

"Exergy loss or destruction of the compressor"

$ED_C = \dot{m}_a * T_0 * (S_2 - S_1)$

"Exergy loss or destruction of the gas turbine"

$ED_{GT} = \dot{m}_a * T_0 * (S_4 - S_3)$

"Exergy loss or destruction in HRB"

$ED_{HRBT} = \dot{m}_a * T_0 * (S_5 - S_4) + \dot{m}_f * T_0 * (S_6 - S_{13})$

"Exergy loss or destruction in the organic turbine-1"

$ED_{OT1} = \dot{m}_f * T_0 * (S_7 - S_6)$

"Exergy loss or destruction in the solar reheater"

$ED_{reheater} = \dot{m}_f * T_0 * (S_8 - S_7) - Q_{s2} * (T_0/T_s)$

"Exergy loss or destruction in the organic turbine-2"

$ED_{OT2} = \dot{m}_f * T_0 * (S_9 - S_8)$

"Exergy Destruction in the regenerator"

$ED_{REG} = \dot{m}_f *T_0 *(S_{10}-S_9) + \dot{m}_f *T_0 *(S_{13}-S_{12})$

"Exergy loss or destruction in the condenser"

$ED_{cond} = \dot{m}_f *T_0 *(S_{10}-S_9) + \dot{m}_f *T_0 *(h_{10}-h_{11})$

"Exergy loss or destruction in the organic Rankine cycle pump"

$ED_{OP} = \dot{m}_f *T_0 *(S_{12}-S_{11})$

"Exergy transfer in the gas turbine cycle by combustion chamber"

$ET_{cc} = Q_{s1} - \dot{m}_a *T_0 *(S_3-S_2)$

"Exergy transfer by the HRB to organic fluid"

$ET_{HRB} = Q_{HRB} - \dot{m}_f *T_0 *(S_6-S_{13})$

"Exergy transfer by the solar heater to organic fluid"

$ET_{reheater} = Q_{s2} - \dot{m}_f *T_0 *(S_8-S_7)$

"Second law efficiency of the gas turbine power plant"

$$\eta_{II\ law\ GTP} = \frac{W_{GTP}}{ET_{CC}}$$

"Second law efficiency of the organic Rankine cycle"

$$\eta_{II\ law\ ORC} = \frac{W_{ORC}}{ET_{HRB}}$$

"Second law efficiency of the combined cycle plant"

$$\eta_{II\ law\ CCP} = \frac{W_{ORC}+W_{GTP}}{ET_{CC}}$$

3.1 Input Parameters

The input parameters taken for computation of results are given below:

Table 1: input parameters for numerical computations for three ORC systems

S.No.	Parameters	Numerical Value
1	Air compressor inlet pressure	P_1= 100 kPa
2	Isentropic efficiency of air compressor	$\dot{\eta}_C$= 0.80
3	Air compressor inlet temperature	T_1= 27˚C
4	Pressure ratio of compressor	r_p= 18
5	Isentropic efficiency of Gas turbine	$\dot{\eta}_{GT}$= 0.85
6	Power output of the Gas Turbine Plant	W_{GTP}= 20 MW
7	Pinch point temperature difference	ΔT_{pinch}= 20˚C
8	Organic Rankine turbine inlet pressure	P_1= 2500 to 4200 kPa
9	Solar irradiation on CSP collector	I_S= 750 W/m^2
10	Isentropic efficiency of organic turbine	$\dot{\eta}_{OT}$= 0.85
11	Effectiveness of regenerator LVHE	e = 0.85
12	Mirror efficiency of CSP collector	$\dot{\eta}_m$= 0.65
13	Efficiency of organic Rankine pump	$\dot{\eta}_{ORP}$= 0.85
14	Condenser Temperature	T_{cond} = 40˚C
15	Dead State Temperature	T_o= 25^{0}C
16	Organic fluid used	Acetone. R134a, R227ea,and R1234yf, R1234ze

4. Results and Discussions

A computational model is developed by using Engineering Equation Solver (Klein and Alvarado, 2005) for evaluating Exergy and the energy analysis of combine cycle plant with Solar reheating and Regeneration. The input data for evaluation are mentioned in chapter 4 except the parameter, whose effect is discussed in particular plot, has been varied.

4.1 Comparison of the various organic fluids

Table-1 shows the comparison variation in first law efficiency of three systems (i.e. (i) the organic Rankine cycle using R134a is which is about 13.96%, (ii) Using R134a in ORC with Regeneration system 15.64 %, and (iii) ORC with Regeneration and solar Reheat system and using R134a is 12.37%) as compared to (i) the organic Rankine cycle using R1234yf is which is about 13.1%, (ii) Using R1234yf in ORC with Regeneration system 14.94%, and (iii) ORC with Regeneration and solar Reheat system and using R1234yf is 11.56%) respectively. Similarly using R1234ze in the three systems the results comes out to be (i) the organic Rankine cycle is which is about 13.09% (ii) ORC with Regeneration system 14.95%, and (iii) ORC with Regeneration and solar Reheat system is 10.23%). It was also observed that the efficiency against the organic turbine inlet pressure and temperature. From the table-1, it is obvious that the efficiency of organic Rankine cycle increases with regeneration while the efficiency of combined cycle plant remains constant. By using the regeneration heat supplied to the organic cycle plant decreases hence the rate of evaporation increase which results output increases and hence the organic Rankine cycle efficiency increases. With solar reheating organic Rankine cycle plant shows increase in efficiency while the combined cycle plant shows decrease in efficiency because of decrease in mean temperature of the heat addition in the system with superheating.

4.2 Comparisons of Exergy efficiency with Regeneration without Regeneration and Regeneration with Solar reheat cycle using different ecofriendly organic fluids

Table-2 shows the effect of various ecofriendly organic fluids on second law performance of various ecofriendly organic fluids in the three systems and it was found that second law efficiency becomes to be43.16% using R134a in organic Rankine cycle and 49.97% using R134a in ORC with Regeneration system and 48.51% ORC with Regeneration and solar Reheat system using organic R134a fluid. Respectively. It is was observed that second law efficiency (in terms of exergetic efficiency) of the organic Rankine cycle increases with maximum pressure and maximum temperature. It is seen that efficiency of combined cycle plants decreases with increase in organic Rankine cycle maximum temperature but increases with increase in the maximum pressure.

4.3 Exergy efficiency

Table-3 shows the thermodynamic performances of ORC with Regeneration and solar Reheat system using different ecofriendly refrigerants at 2500kPa maximum pressure and 120˚C temperature of ORC fluid. It was observed that maximum exergy destruction based on exergy input to be fund using organic R1234ze fluid.

Table-1: Thermal performance (First Law Efficiency) of ORC

System	R134a	R1234yf	R1234ze	R227ea	R407c	R245fa
ORC with Regeneration and solar Reheat system	0.1237	0.1156	0.1023	0.1061	0.1591	0.09879
ORC with Regeneration system	0.1564	0.1494	0..1495	0.1307	0.1940	0.1390
Organic Rankine Cycle	0.1396	0.1310	0.1309	0.1109	0.1532	0.1186

Table-2: Exergetic Efficiency (Second Law Efficiency) of ORC

System	R134a	R1234yf	R-1234ze	R227ea	R245fa
ORC with Regeneration and solar Reheat system	0.4851	0.4833	0.4811	0.4860	0.5038
ORC with Regeneration system	0.4897	0.4910	0.492	0.4908	0.4913
Organic Rankine Cycle(ORC)	0.4316	0.4310	0.4294	0.4290	0.4232

Table-3: Thermal performances of ORC with Regeneration and solar Reheat system

Eco Friendly Heating organic Fluid	R134a	R1234yf	R-1234ze	R-227ea	R245fa
$Eff._{First\_Law}$	0.1237	0.1156	0.1023	0.1061	0.1591
$Eff._{Second\ \_Law}$	0.4851	0.4833	0.4811	0.4860	0.5038
$EDR_{Rational}$	0.5149	0.5267	0.5989	0.5240	0.4962

5. Conclusion

In this paper, an extensive first law (energy) and second law (Exergy) analysis of R134a, R245fa, R1234yf, R1234ze, R227ea organic fluids in combined cycle with regeneration and reheating is presented. Some conclusions of this analysis are summarized as follows:

1. Exegetic efficiency (second law efficiency) and Energy efficiency (first law efficiency) of Organic Rankine cycle with Regeneration is higher than without Regeneration for all selected organic fluids.
2. R1234yf have higher first law efficiency and exergetic efficiency (second law efficiency) in Organic Rankine cycle but R1234ze have higher exergetic efficiency (second law efficiency) improvement from basic system.
3. When solar reheating is done in Organic Rankine cycle than efficiency of Rankine cycle is improved but the efficiency of combined cycle is decreased.
4. Efficiency of combined cycle plant remains almost constant with Regeneration.
5. With increase in maximum pressure of Rankine cycle, efficiency of combined cycle shows increasing trend while with increase in maximum temperature of Rankine cycle, efficiency of combined cycle shows decreasing trend.
6. R1234yfis recommended for practical applications due to its highest exergetic efficiency among selected organic fluid but some important problems related to low flammability and explosion risk have to be considered while managing it.
7. R245fa has highest exergetic efficiency and compared to remaining selected organic fluids and it is recommended for practical applications.

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Nomenclature

Symbols/Subscript

ED	Exergy destruction
Qs1	Heat addition to combustion chamber
Qs2	Heat addition to HRSG
Qs3	Heat transfer to HRB
HRB	Heat recovery boiler
GT	Gas turbine
ST	Steam turbine
OT	Organic turbine
W	work
C	Compressor
P	pump
m	mass flow rate
CC	combustion chamber Efficiency
HRSG	Heat recovery steam generator
ORC	Organic Rankine cycle
a	Air
COND	Condenser