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Thermodynamic analysis of combined cycle power plant with trans-critical cycle integrated with solar system for power generation for space heating and cooling

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Abstract

To build up a programming code and solving it on computational numerical technique by developing energy balance equation for combine cycle power plant integrated with Trans-critical cycle. A computer programming code has been used to simulate a performance under various different conditions under different temperature and pressure circumstances and thermal performance on different cases of Brayton cycle have been analyzed using standard fluids. In this paper Brayton cycle, Rankine cycle and trans-critical cycle are combined which makes it tri-generate cycle and variations of temperature and pressure with thermal performance parameters such as thermal efficiency, exergy destructions in the various components using trans-critical cycle have been analyzed. It is observed that by increasing condenser temperature of Rankine cycle, the combined cycle efficiency , net-work are increasing and by increasing the mass flow rate of Brayton cycle, combined cycle efficiency is decreasing while combined net-work is increasing. Similarly by increasing the exit pressure of pump in the trans-critical cycle, both the combined cycle efficiency & combined net work are decreasing also is decreasing and also by increasing the inlet temperature of turbine of trans-critical cycle, the combined cycle efficiency and combined net-work is increasing. By increasing the inlet pressure of pump of trans-critical cycle, combined cycle efficiency and combined net work (i.e. both) are increasing. .

Keywords: Thermodynamics, Combined cycle power plant, integrated solar system

1. Introduction

The Combined cycle power plants also calls as CCPT are the combination of a gas turbine or Brayton cycle with Rankine cycle. They are designate by the high efficiency in the technology of production of electricity from fossil fuels. Simultaneously, these units are in between to the fast developing currently achieving efficiencies around 60%. Combined cycle power plants defines a number of benefits, listed as speedy construction time, mild investment costs, highly reliable and flexible. The CO_2 emission at the efficiency 0.60 is equal 330 kg--CO_2/MWh, i.e. 2 times lower than the emission from modern coal power plants (800 kgCO_2/MWh). Presently, this technology is not famous because of the high price of gas. The electric efficiency of the Brayton cycle mainly relies on the pressure ratio in the air compressor and the highest temperature in the circuit that is a Combustor Outlet Temperature (COT) of the flue gases from the combustion chamber. However, often because the most vital temperature within the rotary turbine shall be thought of the common exhaust gas temperature at the recess to the turbine, Turbine Recess Temperature (TRT).

Combined cycle power plant with the trans-critical cycle is explained here. With its combination with CO_2 trans-critical cycle it's become tri generative cycle. It has been applied in refrigeration (commercial) and heat pump water heating systems. Its cycle COP (coefficient of performance) mainly depends on high pressure optimization and control. Theoretically, the optimal high pressure increases with the increasing gas cooler outlet temp or ambient temperature. However, due to reliability and cost concerns, there is an upper limit of the high pressure in compressor design and system operation. In this work, i proposed a constrained optimization method for getting constrained optimal high pressure equation of CO_2 trans-critical cycle.

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1.1 Combined cycle power plant

Combined cycle power plant works on two heat engines that is Brayton cycle and Rankine cycle. In order to explain the complete combine cycle power plant one need to explain the Rankine cycle and Brayton cycle separately and trans-critical cycle is also added.

1.2 The Rankine cycle

The Rankine cycle thoroughly describes the way through which team-operated heat engines largely found in thermal power generation plants for generating power. The sources used in these power plants are sometimes fission or the combustion of the fuels accessible. The power of the Rankine cycle is prohibited by the high heat of vaporization of the operating fluid. Also, unless the pressure and temperature reach to their super critical levels within the vessel, the temperature vary the cycle will operate over is sort of small: turbine entry temperatures air usually around 565°C and steam condenser temperatures air around 30°C. This low rotary turbine entry temperature (compared to a rotary turbine) is why the Rankine (steam) cycle is usually used as a bottoming cycle to recover otherwise rejected heat in combined-cycle gas turbine power plant as additionally called gas turbine power stations.

In Rankine cycle working fluid usually circulated in a close loop and keeps recycled at every initial stage. The condensed droplets of water vapour coming out form the Rankine cycle power station through the cooling tower system not actually forms closed loop. In further step addition of steam at high temperature is required for which departure of waste product form the cycle is necessary and all this happens in a closed cycle. Generally, Q_{out} is used to represent the heat coming from the exit point, it is also called as exhaust heat .Latent heat of vaporization of the working fluid is being absorbed by the cooling towers which works as a heat exchangers and simultaneously evaporating cooling water to the atmosphere. There are many substances present which can be used as working fluid but mainly water is considered as a working fluid as it is nontoxic, easily available.

Basically there are totally four process in the working of Rankine cycle which shown below in the T-s diagram of the Rankine cycle in the below parts shown in fig.2.

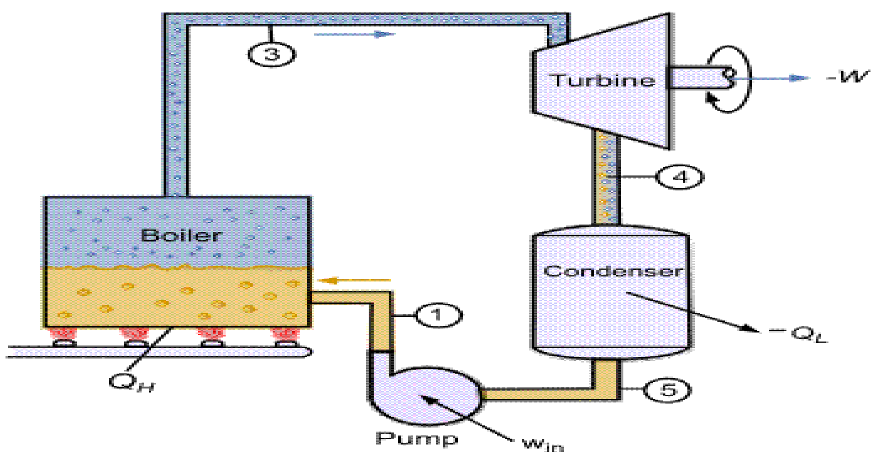

Figure 1: Basic design of Rankine cycle [1].

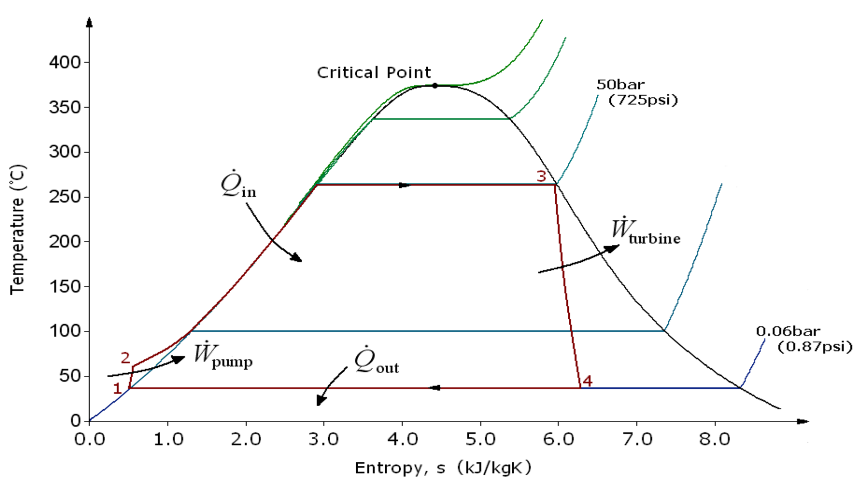

Figure 2: T-s diagram for Rankine cycle.

Process 1-2

In this process a pump is used which is shown in the Fig.2. Here pump is used to produce high pressure to the working fluid so that it can reach up to the boilers.

Process 2-3

As the working fluid reached boiler through pump, it starts getting heated through some external source so as to make working fluid as a dry saturated vapour for calculating required input energy.

Process 3-4

In this process turbine is used where expansion of dry saturated vapour takes place for the generation of power. Occurrence of condensation along with depletion of temperature and pressure is seen. Numerically using the steam tables one can calculate the outputs.

Process 4-1

Here on this process condenser is used where saturated liquid enters the condenser to condense at constant pressure.
In a perfect temperature cycle, the pump and rotary engine will be like physical property, that is no entropy will be generated by either pump or rotary engine and thus maximize output work. On T-s diagram (Fig.2), the vertical lines are used to define the process 1-2 as well as process 3-4 and additionally closely match that of the Carnot cycle. In Fig.2, it is shown that prevention of vapour ending with in the superheat region in temp unit cycle.

1.3 Brayton Cycle

The Brayton cycle is the ideal cycle for gas-turbine engines. It is used for gas turbines in which both the compression and expansion processes are implemented. There are two different types of the Brayton cycle; the open gas turbine cycles and the closed gas turbine cycle respectively. The difference between these two cycles is that during the open gas turbine cycle, a combustion process takes place, and exhaust gases are thrown out, in other words the exhaust gases cannot be recirculates, while in the other cycle (the closed gas turbine cycle), the combustion process is replaced by a heat-addition process, the exhaust gases are also utilized so as to increase the temperature of the air which enters the compressor. In Fig.3, we can see an open gas turbine. Initially, at ambient condition fresh air is taken up by the compressor, and here as a result of compressions the temperature and pressure is increased.. Here, the combustion process occurs at constant temperature. In the third process the high temperature gas enters in to turbine as you can see in Fig.3. In this operation, at atmospheric pressure gases (hot) will expand in the turbine so that power generation can take place. At last exhaust gases coming out from the turbine will move back to the heat exchanger to atmosphere.

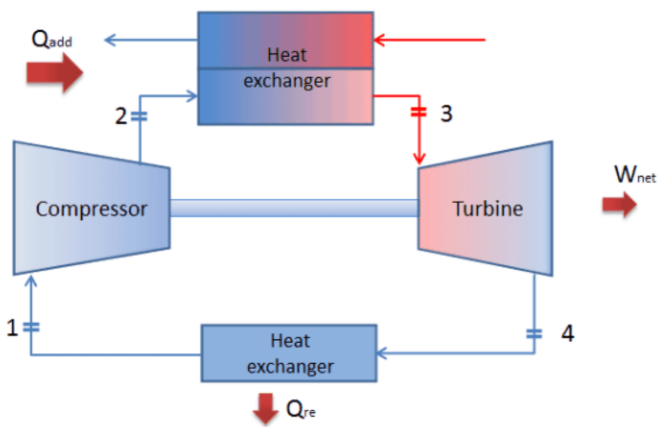

Figure 3: The Brayton cycle [3]

In the shut gas turbine cycle, in spite of the fact that the pressure and development process have in like manner, burning chamber is supplanted by a warmth exchanger in which builds the compacted air temperature. As given in Fig.3, perfect Brayton cycle is really a shut gas turbine cycle, and the means of the Brayton cycle resemble following processes.

Process 1–2

It is an isentropic compression in the compressor, in this process a pump is used which is shown in the Fig.3. Here pump is used to produce high pressure to the working fluid so that it can reach up to the boilers.

Process 2–3

It is known as constant pressure heat addition. As the working fluid reached boiler through pump, it starts getting heated through some external source so as to make working fluid as a dry saturated vapour. Graphically and numerically we can calculate the required input energy.

Process 3–4

It is an isentropic expansion in the turbine. In this process, the turbine is used where expansion of dry saturated vapour takes place for the generation of power and occurrence of condensation along with depletion of temperature and pressure is observed. Numerically using the steam tables one can calculate the outputs.

Process 4–1

It is constant pressure heat rejection, here on this process condenser is used where saturated liquid enters the condenser to condense at constant pressure.

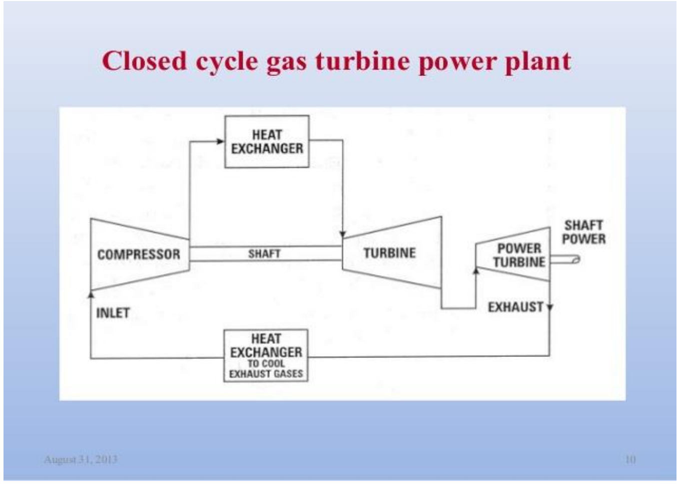

Figure 4: A closed-cycle gas turbine engine [4]

Gaseous petrol as fuel is utilized to run an open Brayton cycle, flammable gas is copied with compacted air to deliver high weight hot vent gases having temperature over 1000°C. These gases extends in the turbine driving the generator and the fumes gases are utilized as a fuel in squander warm recuperation kettle or warmth recuperation steam generator. The superheated steam at that point drives the turbine therefore creating extra power alongside the power delivered from Brayton cycle. Once in a while the turbine from both the cycles is in single shaft and they together drive the generator to deliver power, for the most part the two turbines have their own particular generator. The gas turbine is a standout amongst the most productive one for the change of gas powers to mechanical power or power. The utilization of distillate fluid powers, generally diesel, is likewise normal as exchange energizes. All the more as of late, as straightforward cycle efficiencies have enhanced and as petroleum gas costs have fallen, gas turbines have been all the more generally embraced for base load control age, particularly in consolidated cycle mode, where squander warm is recouped in squander warm boilers, and the steam used to deliver extra power. They are installed around the world. Consolidated cycle control plant as in name recommends, it joins existing gas and steam innovations into one unit, yielding noteworthy changes in warm proficiency over traditional steam plant. Anyway the warmth recouped in this procedure is adequate to drive a steam turbine with an electrical yield of roughly 50 for every penny of the gas turbine generator. The gas turbine and steam turbine are coupled to a solitary generator. For start-up, or 'open cycle' task of the gas turbine alone, the steam turbine can be detached utilizing a pressure driven grasp. As far as general speculation a solitary shaft framework is regularly around 5 for every penny bring down in cost, with its working straightforwardness ordinarily prompting higher unwavering.

1.4 Carbon dioxide Trans-critical Power Cycle

The current study conjointly focuses on the trans-critical cycle due to its high potential usage within the trade and thanks to the restricted studies found within the literature. The trans-critical cycle, whose heat rejection takes place at a subcritical pressure, should not be confused with the entirely critical cycle.

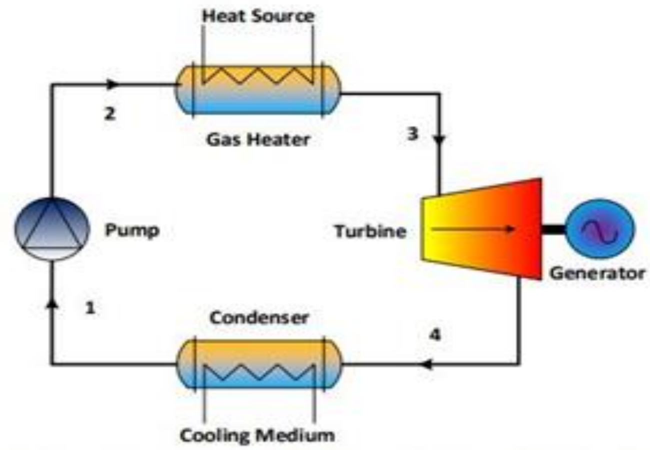

Figure 5: The trans-critical cycle [5]

Actually, coal pink-slipped trans-critical power plants at high temperatures (above five hundred °C) represent a mature technology and among the most effective playing heat engines with a thermal potency as high as forty ninth .As way because it is understood the dioxide are going to be thought-about as trans-critical cycle wherever the temperature is on top of vital temperature i.e., 31 °C. For trans-critical carbon dioxide because it is portrayed in Fig.6 and therefore the T-S diagram in Fig.6 same as Brayton cycle the trans-critical dioxide cycle can expertise processes: compression (1-2), isobaric heat provide (2-3), growth (3-4), and isobaric heat rejection (4-5). The sole distinction between these 2 cycles is whether or not a part of the cycle is found within the subcritical region or not. Therefore, each cycle' generally associated with critical cycles within the literature.

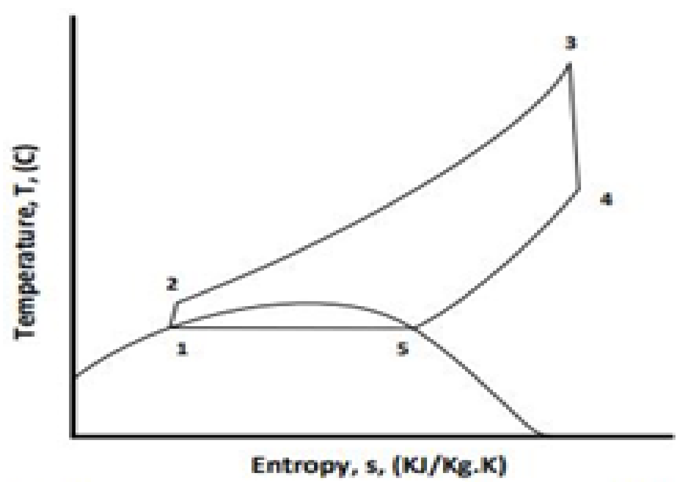

Figure 6: T-S Diagram of Trans-critical Cycle

The energy analysis relies on the primary law of physics. The thermal potency and therefore the specific internet output ar its results. With the assumptions antecedent explicit, their values rely solely on one freelance parameter: the air mass, that ar P2=P3. Above all, these results don't depend upon the operating fluid mass rate. The equations for the various parts. For the pump: $\eta_p = (h_{2,s}-h_1)/(h_2-h_1)$ wherever the η_p is that the potency of the pump, h_1 is that the specific physical property of the pump recess fluid, h_2 is that the physical property of pump outlet fluid and $h_{2,s}$ is that the physical property physical property of outlet fluid. $w_p = h_2 - h_1$ wherever the w_p is that the work of the pump, $h_1$is the particular physical property of the recess fluid and h2 is that the physical property of outlet fluid. For the turbine: $\eta_t = (h_3-h_4)/(h_3-h_{4,s})$ wherever the η_t is that the potency of the rotary engine, h3is the particular physical property of carbon dioxide at the rotary engine recess, h_4 is that the physical property of outlet fluid and h4,is is that the physical property physical property of outlet fluid. $w_t = h_3 - h_4$ wherever the w_t is that the work of the pump, $h_3$is the particular physical property of the rotary engine recess fluid and h_4 is that the specific physical property of outlet fluid. For the vapor generator: $q_{in} = h_3 - h_2$ wherever the q_{in} is that the heat transferred to the fluid in vapor generator, h_2 is that the specific physical property of the warmer recess fluid and h_3 is that the physical property of outlet fluid. For the condenser: $q_{out} = h_4 - h_1$ wherever the q_{out} is that the heat rejected from the operating fluid in condenser, h4is the physical property of the fluid coming into the condenser and h1 is that the physical property exiting of fluid deed the condenser. The thermal potency of the cycle:

$$\eta_{th} = [(W_t-W_p)/q_{in}] = [(h_3-h_4)-(h_2-h_1)/(h_3-h_2)]$$

wherever the η_{th} is that the thermal potency of the cycle, w_t is that the rotary engine work, W_p is that the pump work and q_{in} is heat transferred to the operating fluid in warmer. In compare with associate degree organic Jeannette Rankin cycle (ORC), the carbon dioxide trans-critical power cycle encompasses a higher capability in taking blessings of the energy during an inferior waste heat with gradient temperature, like exhaust gases. The temperature glide (Temperature modification throughout take-up of warmth energy) for carbon dioxide on top of the crisis permits for an improved.

1.5 Turbine Cycle

An ignited mixture of a natural gas and air which is purified and a later on compressed makes it to expand. Spinning of a turbine blades took place due to a pressure created form expansion process, for creating electricity ay are attached with a generator as well as shaft. In further second step a generation of steam is done by utilizing a heat of gas turbine from exhaust and make it to through a heat recovery steam generator (HRSG).

1.6 Heat Recovery Steam Generator

In HRSG that stands for Heat Recovery Steam Generator is known for water flow in a tube and a flow is extremely purified and around am passing of a hot gases occurs and therefore ay produce steam .In a steam turbine rotation of steam occurs and concatenate with a generator for a production of Electricity.A steam condensing and water system is a same as in a steam power plant.

1.7 Typical Size and Configuration of CCGT Plants

Rankine and Brayton cycle combines to form a combined-cycle. It includes different configuration for shafts such as multi shaft configuration and single shaft configuration. Single-shaft systems constitute of one - one unit of each gas turbine, steam turbine, generator and Heat Recovery Steam Generator (HRSG), with a gas turbine and steam turbine concatenate to a single shaft of a single generator. Areare one or more generator in multi turbines and HRSGs to a separate single steam turbine-generator steam is supplied through it.
A chief demerit of multiple stage combined cycle power plant is that a number of steam turbines, condensers and condensate systems-and perhaps a cooling towers and circulating water systems increases to equalise a quantity of gas turbines.

1.8 Efficiency of CCGT Plant

It is essential to use a parts of a exhaust energy through gas to gas healer after a combination of Rankine cycle and a bray ton cycle, achievement of low temperature at output and high temperature at a input can be done. A efficiency of a cycles will sum up with it, as same fuel source powered am. A power of CCPT is filled by Supplementary Firing and Blade Cooling. At HRSG supplementary firing was organised and a flow of compressed gas flow is being carried out in a turbine area of a rotary engine and it was mainly opted for a cooling down of a turbine blades.

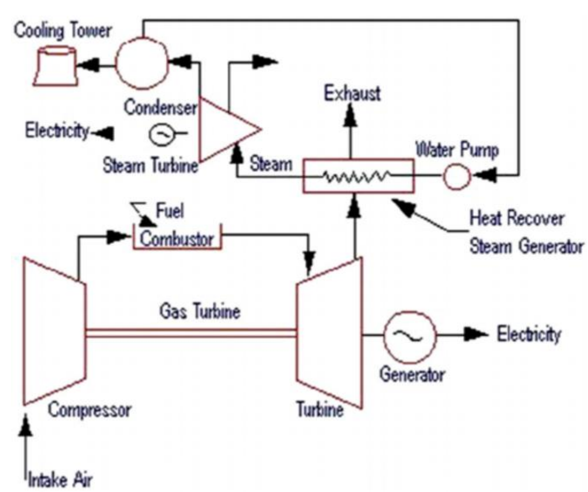

Figure 7: Combined Cycle Gas Turbine Plant HRSG [6]

1.9 Combined cycle integrated with solar system

PTS called as a parabolic trough solar energy plant. Ay are a most vital establishment of a CSP that is concentrating solar energy technique. SEGS that is PT star electricity producing technique system shows that a tendency of this present technique found to be more trust worthy as well as renewable resource. This method usually adopted on a commercial scale on large scale with some around output of 345MW. For satisfying a increasing demand for a energy resource, CSP are found to be a most promising technology. It has been seen that only CSP can fulfil a demand of energy for developing countries. Electricity produce by a CSP with in a MENA (Mediterranean & geographical region) wants to improvise a native energy production system. Trans Mediterranean renewable energy cooperation introduced a energy cooperation. ISCC is one of a most important system. It stands for integrated solar combine cycle; PTT that is parabolic trough technology is said to be coupled whenever it is supported by a star field. Are is no need to install a storage source for a backup of fuel for boilers because of a flexibility in operation and reduction in value of a system. Here a mathematical codes are built for a integrated solar combine cycle that is ISCC powerhouse operational below weaar conditions. Habib (1994)[1] presented an analysis of a cogeneration system, which quantified the irreversibility of the different components of each plant. In addition to that, the influence of the heat-to-power ratio and the process pressure on the thermal efficiency and utilization factor is presented. His results show that the total irreversibility of the cogeneration plant is 38 percent lower compared to the conventional plant. This reduction in the irreversibility is accompanied by an increase in the thermal efficiency and utilization factor by 25 and 24 percent, respectively. The results show that the exergy destruction in the boiler is the highest. Bejan, et al (1996)[2] actually provided a comprehensive and rigorous introduction to thermal system design and optimization from a contemporary perspective. The book includes current developments in engineering thermodynamics, heat transfer, and engineering economics relevant to design. The use of exergy analysis and entropy generation minimization is featured. A detailed description of engineering economics and thermo economics are also presented. Moreover, a case study is considered throughout the book for continuity of the presentation. The case study involves the design of a gas turbine cogeneration system. Rolf Kehlhofer (1997) [3] provided the study of thermodynamic principles of combined cycle power plants and co-generation system. His work includes the different layout system of combined cycle power plants. It describes the effect of operating variables and part load behavior of combined cycle power plants. Karthikeyan et al (1998) [4] derived energy balances for a one pressure level heat recovery steam generator. Effects of pinch and approach points on steam generation and also on temperature profiles across heat recovery steam generator was investigated. The effects of operating conditions on steam production and also on exit gas temperature from the heat recovery steam generator are discussed. It is concluded that low pinch point results in improved heat recovery steam generator performance due to reduced irreversibility's. Additionally, the supplementary firing enhances the steam production. Tawney, et al. (2000)[5] focused on several ranges of process steam flows and conditions in order to provide a basis for comparison of the most common cycle configurations in combined cycle applications. Plant design, cycle performance, and economics of each configuration are evaluated based on requirements of flexibility and process steam flows. Rather than self-establishing the energy balances, Gate Cycle TM Heat Balance software developed by GE Enter Software, Inc. is used to build thermal models. Additionally, a financial software tool developed within Bechtel is used to construct an economic model for each cycle configuration. It is concluded that, the selection of a cogeneration facility type and the economic parameters are very much site specific and are based on numerous variables such as site ambient conditions, the level of desired power output and steam demand, capacity factor, flexibility, power purchase agreement and steam purchase agreement requirements, and owner's economic parameters for return on equity. Boyce (2002) [6] covered all major aspects of power plant design, operation, and maintenance. It covers cycle optimization and reliability, technical details on sizing, plant layout, fuel selection, types of drives, and performance characteristics of all major components in a cogeneration or combined cycle power plant. Comparison of various energy systems, latest cycles and power augmentation techniques, reviews and benefits of latest codes, detailed analysis of available equipment, techniques for improving plant reliability and maintainability, testing and plant evaluation techniques, and advantages and disadvantages of fuels. ET. Bonataki and K.C. Gianna Koglou (2002) [7], gave a modern optimization methods based on evolutionary algorithms and game theory. They will be supported by computational methods for their thermal analysis and simple model for computing there capital cost. In this paper a detailed thermal model and a quite simple economic model will be incorporated in an evolutionary search algorithm and an automated tool combine the design of new cycle gas turbine power plant will be demonstrated. The search algorithm is used a Genetic Algorithm (GA) for single and multi-objective optimization problems. The latter is handled through game theory inspired enhancement to the G.A. based search yielding the so called optimal Pareto front, with two objectives (via electrical efficiency and investment cost), the Pareto front members can be envisaged as compromise between high cost/high efficiency and low cost/low efficiency design. Neil Petchers (2003) [8] provided a comprehensive details of thermal system design from a contemporary perspective. First part of the book provide a theoretical basis for understanding the inter relation of heat and power resources. It 18 provides and introduction to basic heat and power thermodynamic and includes sections on heat and power generation technologies and equipment. Part seconds describe the infrastructure in which the theory and technologies describe in part first must be applied. Having learned on the theory and available technology, application cannot be effective device, analysis' for cost effectiveness and implemented without knowledge of environmental factors and utility rate structures. Yongjun Zhao, et al (2003) [9], investigated the design and cost of HRSG system and to demonstrate impact on the overall cost of electricity (COE) of a combined cycle power plant. There are numerous design parameter that can affect the size and complexity of the HRSG, and it is the plan for the project to identity all the important parameter and to evaluate each the exhaust gas pressure drop across the HRSG is chosen for evaluation. This parameter affects the performance of both the gas turbine and steam turbine and size of HRSG. Single pressure, two pressure, and there pressure HRSG are investigation with the trade-offs between design point size, performance and cost evaluated for each system. A genetic algorithm is used in the design optimization process to minimize the investment cost of the HRSG second system level metrics' are employed to evaluate a design. They are gas turbine net power, steam turbine net power, fuel consumption of the power plant, net cycle efficiency of the power plant, HRSG investment cost, total investment cost of the power plant and the operating cost measured by the cost of electricity (COE), The impacts of HRSG exhaust gas pressure drop and system complexity on these system level metrics are investigated. Zaleta-Aguilar Alejandro (2003)[10],represented the proposed exergo-economic fuel-impact models for steam turbines in power plants. They are applied to calculate the impact on the steam cycle when malfunctions are occurring during the operation of steam turbine sections. Concepts such as the exergetic consumption and the dissipation temperature are used to understand the proposed fuel-impact analysis. In order to validate these fuel-impact methods, well-known procedures, to simulate on and off-design conditions of a steam power cycle are used as references. Chih Wu, (2004) [11] provided an intelligence computer software called "cycle pad". It is powerful, mature, user friendly package developed to simulate thermodynamic devices and cycles. It makes feasible for engineers to run meaningful sensitivity analysis to consider combinations of design modifications to make engineering cost benefit analysis and to include refinements such as accounting for pressure changes and heat transfers occurring between major cycles components. Yadav, et al (2004)[12] mainly focused on to development of gas turbine related power plants such as combined 20 Mitre, et al (2005), In his paper, author evaluates the effect of operational conditions on pollutants (co, co2, so2, No) emissions levels, waste heat and waste water of a combined cycle natural gas and steam power plants. The HYSYS process simulation was used for modeling and simulation. This study clearly shown that the absolute quantity of pollutants emitted is high. Also it was possible to verify that the unit operation in the condition of minimal emissions regarding the maximum possible, and thus a reduction or elimination of such pollutants is not possible. It can be observed from this study that the ideal condition for exergy productivity is to operate with a fuel air ratio as the stoichiometric one. The first constraints to this ideal is the mechanical conditions of the turbine, which can be operate at the corresponding combustion gas exit temperature so a stoichiometric ratio in the range of (2.7-2.9) is used, and these conditions make the process viable (turbine viability) and minimize pollutants production (CO and NOx). These operational conditions are the optimal considering environmental concerns. The CO2 being a product, is maximized in the process, so there is no need to search for methodologies to minimize their production, but there is for technologies for their capture and uses parallel to the process. Xiaojun Shi and DefuChe (2007)[13]proposed an improved Liquefied Natural Gas (LNG), fuelled combined cycle power plant with a waste heat recovery and utilization system. The proposed combined cycle, which provide power output and thermal energy, consist of gas/steam combined cycle, the subsystem utilization the latent heat of spent steam from the steam turbine to vaporize LNG, the sub system that recovers both the sensible heat and latent heat of water vapour in the exhaust gas from heat recovery steam generator (HRSG), by installing a condensing heat exchanger, and the HRSG waste heat utilization sub system. The conventional combined cycle and proposed combined cycle are modeled, considering mass, energy and exergy balances for every component and both energy and exergy analysis are conducted. Parametric analysis are performed for the proposed combined cycle to evaluate the effect of several factors, such as the gas turbine temperature (TIT), the condenser pressure, the pinch point temperature different of the condensing heat exchanger and fuel gas heating temperature on the performance of the proposed combined cycle through simulation calculation. The results show that the net electrical efficiency and the exergy efficiency of proposed combined cycle can be increased by 1.6% and 2.84% than those of the conventional combined cycle 21 respectively. The heat recovery per kg of flue gas is equal to 86.27 kJ/sec. one MW of electric power for operating sea water pumps can be saved. The net electric efficiency and heat recovery ratio increase as the condenser pressure decreases. The higher heat recovery from HRSG exit flue gas is achieve at higher gas TIT and at lower pinch point temperature of condensing heat exchanger. Srinivas, et al (2008) [14].studied the optimum configuration for single pressure (SP), dual pressure (DP) and triple pressure (TP) heat steam generator (HRSG) to improve heat recovery and exergy efficiency of combined cycle. Deaerator was added to enhance efficiency and remove dissolve gases in feed water. A new method was introduced to evaluate low pressure (LP) and intermediate pressure (IP) in HRSG from local flue gas temperature to get minimum possible temperature difference in heaters instead of a usual fixation of pressures. Optimum location for deaeraor was found at 1, 3, and 5 bar respectively for SP, DP and TP in heat recovery at a high pressure (HP) of 200 bar. It is concluded that optimum pressure ratio for compressor with SP, DP and TP effects in heat recovery are 8, 10 and 12 respectively at 12000 C of gas turbine inlet temperature optimum deaerator pressure is obtained at 1.3, and 5 bar for SP, DP and TP levels respectively at steam turbine inlet pressure of 200 bar. Similarly at 200 bar of HP pressure for DP and TP, steam reheated demands 100 bar to maximize exergy efficiency for combustion chamber. Parametric analysis exhibits that gain in efficiency from single pressure heat recovery to DP and TP recovery increasing with diminishing rate. J.Li, G.Pei, Y.Z.Li, J.Ji[15] find out that with the reducing of the Organic Rankine Cycle (ORC), the engine shaft power isn't solely determined by the enthalpy drop by the expansion method however additionally the external heat loss from the expander. Theoretical and experimental support in evaluating tiny scale expander heat loss is rare. This paper presents a quantitative study on the convection, radiation and conductivity heat transfer from a kW-scale expander. A mathematical model is constructed and valid. The results show that the external radiative or convective heat loss constant was regarding 3.2 or 7.0 W/m2K once the ORC operated around 100ºC. Radiative and convective heat loss coefficients enhanced because the expander operation temperature enhanced. Conductive heat loss because of the affiliation between the expander and also the support accounted for an oversized proportion of the warmth loss. The fitting relationships between heat loss and mean temperature distinction were established. It's recommended that low conduction material be embodied within the support of expander. Mattress insulation for compact expander may be eliminated once the operation temperature is around 100ºC. V. Lemort et al. [16] had presented the results of an experimental study administered on a model of an open-drive oil-free scroll expander integrated into an ORC operating with refrigerant HCFC-123. By exploiting the expander performance measurements, the eight parameters of a scroll expander semi-empirical model are then known. The model is ready to calculate variables of 1st importance like the mass rate, the delivered shaft power and also the discharge temperature, and secondary variables like the availability heating-up, the exhaust cooling-down, the close losses, the interior escape and also the mechanical losses. The utmost deviation between the predictions by the model and also the measurements is two hundredth for the mass rate, 5-hitter for the shaft power and 3K for the discharge temperature. The valid model of the expander is finally wont to quantify the various losses and to point how the planning of the expander can be altered to realize higher performances. Kosmadakis et al. [17] suggested the constant theoretical study of a 2 stage solar Rankine cycle for Ro distillation. The current work issues the constant study of an autonomous 2 stage solar ORC for Ro desalinization. The most aim is to estimate the efficiency similarly on calculate annual energy out there for desalinization. Aleksandra & Borsukiewicz[18]suggested pumping within Rankine cycle and created calculation based mostly result for the pumping work on ORC. Analysis has been administrated for eighteen completely different organic fluids that may be used an operating fluids within the subcritical ORC system. A trial was created to search out correlations between numerous thermo-physical properties of operating fluids, specific work and power of cycle. Kim, Y.M., C.G. Kim, and D. Favrat[19]Every year, the sun irradiates the landmasses on earth with the equivalent of 19,000 billion tons of oil equivalent (toe). Only a fraction (9 billion toe) would satisfy the world's current energy requirements. Put differently, in 20 minutes, the amount of solar energy falling on the earth could power the planet for one year. Chen, H., [20] Solar ponds are large-scale solar thermal energy collectors, which are pools filled with saltwater with a density gradient from the bottom to the top. A solar pond combines heat collection and storage. With a 20°C ambient temperature, the thermal energy obtained from the solar ponds is in the form of low-grade heat at 70 to 80°C. There are low-, medium- and high- temperature solar thermal collectors, depending on their collecting temperature.

2. System Description

2.1 Solar thermal power plants

A sun unendingly provides a vast quantity of energy. As a result of a character of this energy, which is opened up, it has to be collected and targeted to be useable. Are square measure several applications and techniques wherever solar power is utilized. In star thermal power plants, solar power is absorbed as heat that is a re-modelled into electricity. Re-modelling thermal solar power to electricity may be conducted by completely different approaches. A foremost common techniques square measure concentrating alternative energy (CSP) plants and also a star chimney. CSP techniques are: star tower, parabolic dish and parabolic trough. With a star chimney, a radiation is born-again to mechanical energy by heating a air in an air solar dish (greenhouse). A heated air is allowed to flow through a chimney settled at a center of a solar dish. A buoyancy force of a air causes flow through a chimney. A flowing air drives a rotary engine that is fastened at a doorway of a chimney to come up with electricity. A star chimney consists of a solar dish or greenhouse, high created chimney and rotary engine. A storage system will be used mistreatment this method to stay a plant engaging at night-time. A straightforward idea of its storage system is to fabricate water storage to a lower place a absorbent material plate of a star collectors. This technology blessings are; it makes use of beam and diffuse radiation therefore it's able to work throughout cloudy periods, it will work twenty four hours if a storage system is utilized, a desired materials to construct it straightforward and offered in most regions of a globe, and Are’s no want for cooling water systems, therefore it's appropriate for arid locations.

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2.2 Concentrating Solar Power (CSP) plants

CSP plants provide high temperature energy that’s employed to work standard potency cycles like a Stirling engine. Though solar power concentrated plant square measure utilized largely for production of electricity, ay could be employed in several industrial exercise. Fig.8 depicts various practice for CSP systems. One in every foremost necessary conditions for selecting a foremost appropriate method for planned exercise is that an operative temperature.

Figure 8: CSP applications India [9]

2.3 Solar Tower

This method gives a boosted magnitude relation of radiation concentration that permits star towers to realize that for heating of an air high temp is required. Depicted in a Fig. 9. The system of tower consisting a reflectors of heliostat settled in spherical array round a star adopter. Position of a sun being tracked by a reflector to make sure leading a daylight to a adopter. Medium of transfer of heat is employed within aadopter for soak up a focused alternative energy. Rejected heat is provided to work up aarmal power station. Warmth fluid transfer within a central adopter may be air &wateer, melted salt or oils. Analysis releases this method may b accustomed workup of turbine wherever air is pressurized initial so het up within a receiver to a thousand. star tower among a evidenced CSP technologies within a field.. Air capability is ten MWe every

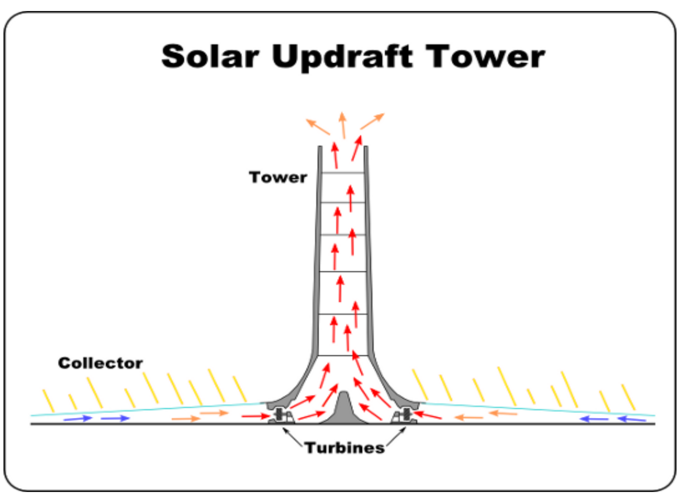

Figure 9: Solar draft (solar tower)[24]

2.4 Parabolic dish-engine

Ground level construction such method is about usage of dish having parabolic shape to focus radiations on a generator of engine within center of attention of a reflector. Under consideration of potency, umbrella dish is that a best technology among star techniques, its highest potency may be a maximum amount of twenty ninth. As you can see in fig.10. A standard umbrella dish having radius variation from five to fifteen m with five to twenty five powers. It is appropriate for redistributed potency provide and unpopulated area by taking this technique.

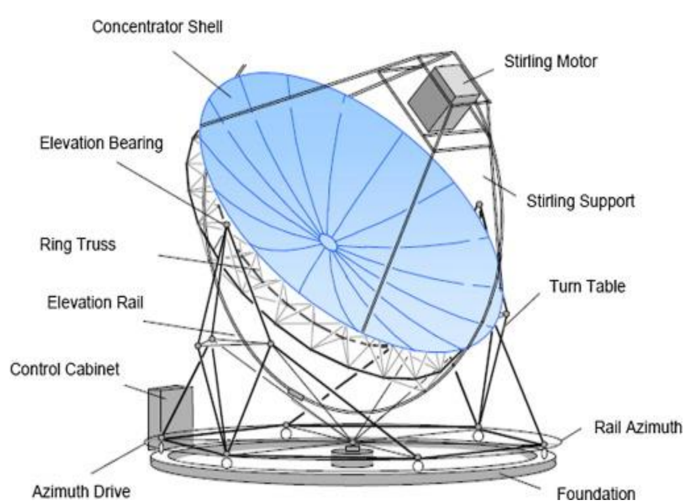

Figure 10: Parabolic dish-collector [25]

2.5 Liner Fresnel system

Setup actually have associate array of liner reflectors. A absorbent tube that is homeward on a focused radiation being received by the focal line of the reflector and converts a sun potency to heat. Fig.11 depicts Augustin Jean Fresnel system parts. H T fluid is employed to soak up this energy to be employed in a planned application. This sort of collector offers sensible prospects for solar power use and it's appropriate for small- and large-scale applications. Some prototypes are tested. Its operation temperature was two hundred °C, its dimensions were sixteen m long × four m high and it consisted of eleven primary reflectors.

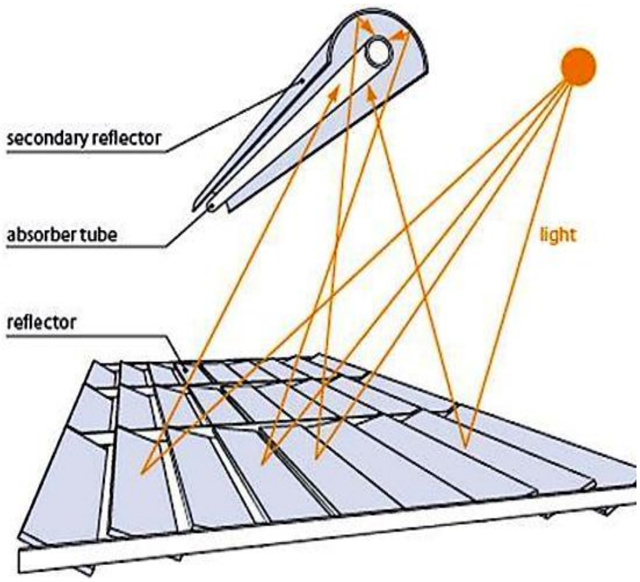

Figure 11: Linear Fresnel system [26]

2.6 Parabolic trough system

A distinction among this technique and hence a Liner Augustine Jean Fresnel system is parabolic trough system utilizes umbrella shaped formed reflectors. A concentration ratio will around eighty or additional .In absorber tube collected energy oh HT fluid will run. Parabolic trough technique provides energy at a temperature of up to 400°C. This energy is provided to run either a straightforward Jeannette Rankin cycle or hybrid system. A warmth transfer fluid that is employed to soak up warmth will be either water or artificial oils. Fig.12 depicts parabolic trough system components.

2.7 Parabolic trough solar power plants

Under this topic an operation eventualities &hence a totally different installation configuration for parabolic trough systems square measure described. Additionally, as this technique utilizes a sun following system, a used following system is in brief mentioned.

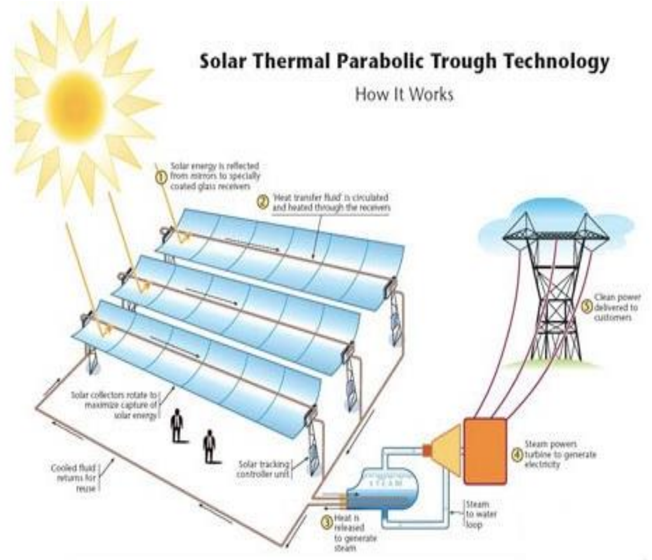

Figure 12: Parabolic trough system [27]

2.8 The sun chase system

Homeward-bound rotational axis, wherever collectors area unit. A system unceasingly drives collectors from east at sunrise to west at sunset. Little motors area unit accustomed drive this chase system. Fig.13 depicts solar furnace system concept. Since solely direct radiation is focused PTS utilizes a sun chase system for confirm most potency of a concentrating method

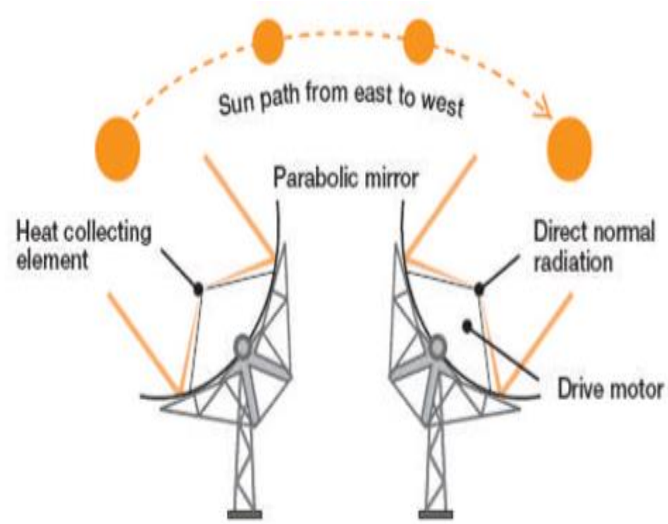

Figure 13: The sun chase system [28]

2.9 Solar desalinization

A potential of victimization star PTS in H_2O desalinization, wherever a star field is attached to water desalinization like multi-stage flash distillation (MSF) or multi impact distillation. Below Fig. 14 depicts solar desalinization.

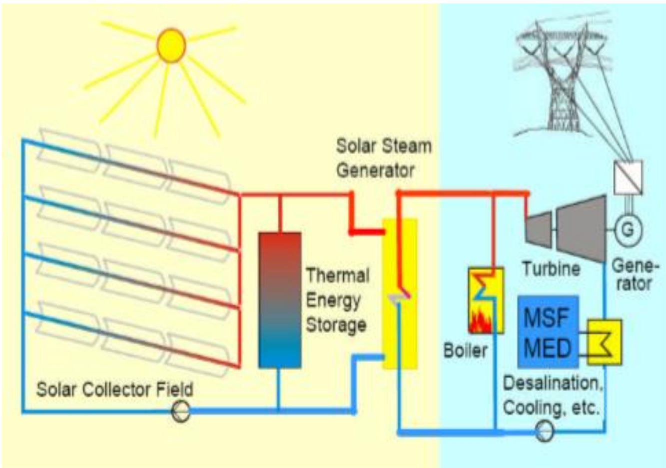

Figure 14: Solar desalinization [29]

2.10 Integrated solar combined cycle

An ISCC system may be a combination of a star field and gas turbine-combined cycle. A waste heat from a turbine is employed to come up with some steam to be distended during a turbine. Additionally, a star field provides further heat to a thermal cycle. An extra heat from a star field ends up in electricity generation increase throughout daylight time. This mixture ends up in up a thermal potency. A advantages of using this technology are to beat some issues associated with start-up and close up in alternative energy plants, scale back a cost of capital and improve a solar-to-electricity potency. Below Fig. 15 shows ISCC.

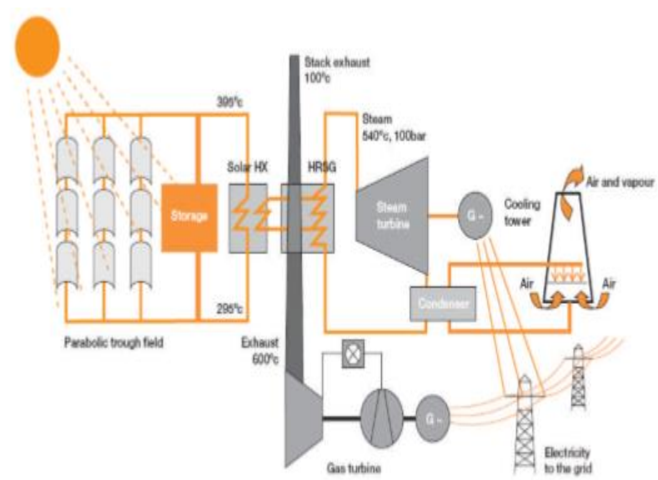

Figure 15: Integrated solar combined cycle [30]

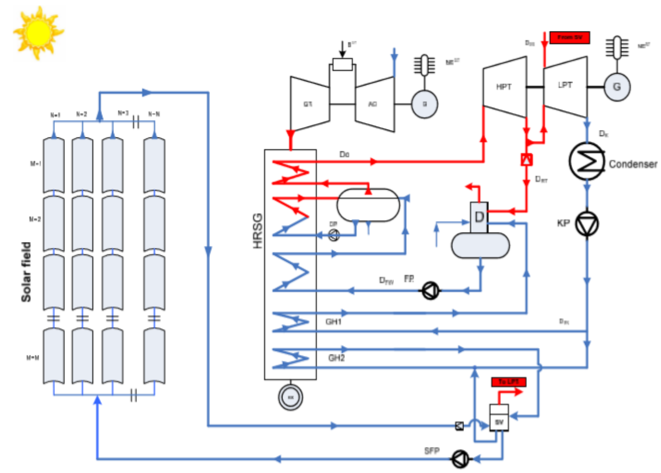

Figure 16: The proposed design for ISCC [31]

2.11 The proposed design

A planned style of a ISCC is shown in Fig.16. Its associate integration between a traditional combined cycle station (gas & steam turbine) and star field, supported a parabolic trough solar furnace. HRSG is one in every of a CC parts. It's wont to recover a warmth loss from a turbine exhaust gases. Most advanced electricity generation gas turbines area unit capable of being connected to heat recovery units. A most parts of a planned ISCC are: turbine unit, HRSG, turbine unit, and star field supported parabolic trough technology.
A turbine unit is that a major energy resource for a political leader cycle. HRSG that could be a device won't to recover heats from high temp. Gases streams (used with gas turbines). The HRSG consists 3 main sections, i.e. superheating section, evaporator & saver. The saver will pump up the feed water temp. To the saturation temperature to recover the max amount heat as potential from the gases stream. Then the steam generator (evaporator) converts the feed water to saturated steam at the HRSG drum's pressure. The superheating section will increase the steam temperature to the required temperature (HRSG, 2007). The projected style includes victimization 2 gas heaters. The primary warmer GH1 preheats water within the HRSG. Additionally an evaporator is employed within the HRSG to come up with some steam to produce some energy

3. Mathematical analysis of the integrated solar combined cycle

An approach wont to investigate various parts of a ISCC powerhouse is described. It includes a physics analysis of main objectives.

3.1 Combined cycle thermodynamics analysis

In an ISCC cycle a sole objective of analyzing a gas turbine unit. It is about to enumerate the energy's wastage in terms of gases at the exit. Under this section we will try to examine the performances of the cc that is combustion chamber,

performance of turbine, and last, compressor.
The equation for heater/heat exchanger of Brayton and Rankine cycle:

$Q_{hex} = \dot{m}_b(h_{4,b} - h_{1,b}) = \dot{m}_r(h_{1,r} - h_{4,r})$ (1)

The equation of the expander of Brayton cycle:

$n_{T,s} = h_2 - h_1 / h_3 - h_1$ (2)

$W_T = \eta_T * (h_1 - h_2)$ (3)

Where η_T is the mechanical efficiency of expander.
The equation of condenser of trans-critical cycle:

$Q_{cond} = \dot{m}(h_4 - h_1)$ (4)

The equation of fluid pump of Rankine cycle:

$W_{pump} = h_4 - h_3$ (5)

The work of turbine of trans-critical cycle.

$W_T = h_3 - h_4$ (6)

The work of turbine of Rankine cycle

$W_T = h_{1,r} - h_{2,r}$ (7)

The work of pump of trans-critical cycle.

$W_C = h_2 - h_1$ (8)

The work of pump of Rankine cycle

$W_P = h_{4,r} - h_{3,r}$ (9)

Equation for the net system output:

$W_{sys} = W_{NET,Rankine} + W_{NET,Brayton} + W_{NET,transcritical}$ (10)

Equation for finding the cycle thermal efficiency:

$\eta_{th} = W_{sys}/Q_{hex}$ (11)

3.2 Exergy Analysis

Economizer

$E_{rev,EC,b} = m_{,dot,b} * ((h_{bv} - h_{5,b}) - T_{amb} * (s_{bv} - s_{5,b}))$

$E_{rev,EC,r} = m_{dot,r} * ((h_{4,r} - h_{rv}) - T_{amb} * (s_{4r} - s_{rv}))$

$E_{rev,EC,total} = E_{rev,EC,b} + E_{rev,EC,r}$

(Total exergy in Economizer)

Heater

$E_{rev,HE,b} = m\_{dot\_b} * ((h_{bs} - h_{bv}) - T_{amb} * (s_{bs} - s_{bv}))$

$E_{rev,HE,r} = m_{dot,r} * ((h_{rv} - h_{rs}) - T_{amb} * (s_{rv} - s_{rs}))$

$E_{rev,HE,total} = E_{rev,HE,b} + E_{rev,HE,r}$

(Total exergy in Heater)

Super heater

$E_{rev,SH,b} = m\_{dot,b} * ((h_{4,b} - h_{bs}) - T_{amb} * (s_{4,b} - s_{bs}))$

$E_{rev,SH,r} = m_{dot,r} * ((h_{rs} - h_{1,r}) - T_{amb} * (s_{rs} - s_{1,r}))$

$E_{rev,SH,total} = E_{rev,SH,b} + E_{rev,SH,r}$

(Total exergy in Super Heater)

Overall hex

$E_{rev,HEX,b} = m\_{dot,b} * ((h_{4,b} - h_{5,b}) - T_{amb} * (s_{4,b} - s_{5,b}))$

$E_{rev,HEX,r} = m_{dot,r} * ((h_{4,r} - h_{1,r}) - T_{amb} * (s_{4,r} - s_{1,r}))$

The exergy in Overall Heat Exchanger is given by following expression

$E_{rev,HEX,total} = E_{rev,HEX,b} + E_{rev,HEX}$

4. Result and Discussion

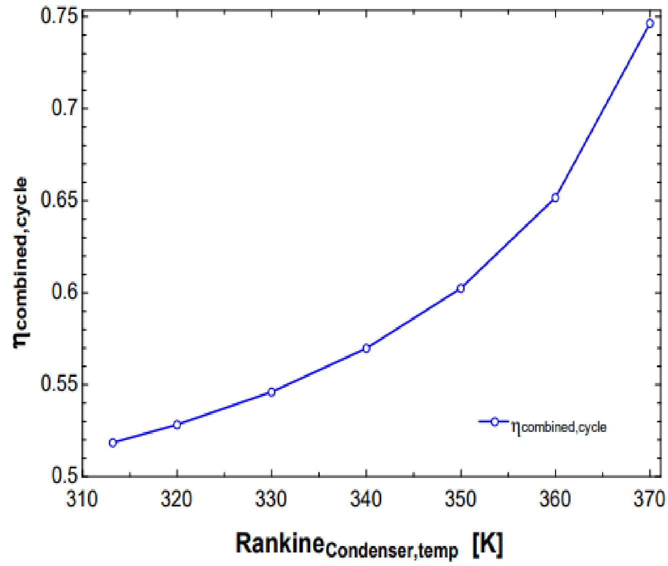

Figure 17: Variation between combined cycle efficiency and condenser temperature of Rankine Cycle

Fig. 17 shows the variation between combined cycle efficiency and condenser temperature of Rankine Cycle. It can be seen that on increasing the condenser temperature, efficiency is also increasing. The combined cycle efficiency is 56.98% at condenser temperature 340K.

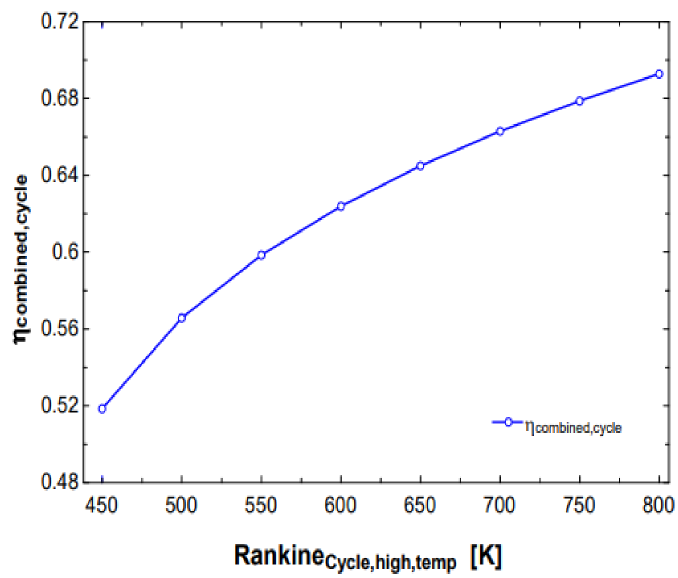

Figure 18: Variation between combined cycle efficiency and cycle high temperature of Rankine Cycle

Fig.18 shows the variation between combined cycle efficiency and cycle high temperature of Rankine Cycle. It can be seen that on increasing the cycle high temperature, efficiency is also increasing. The combined cycle efficiency is 51.85% at temperature 450K.

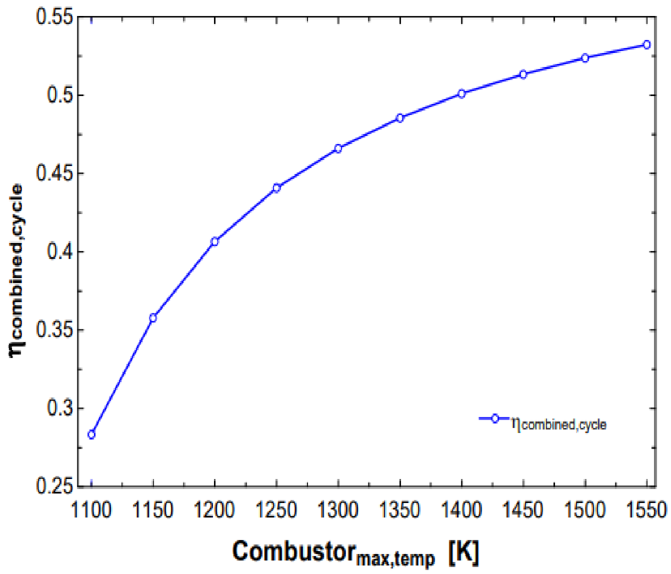

Figure 19: Variation between combined cycle efficiency and combustor maximum temperature of Brayton Cycle

Fig.19 shows the variation between combined cycle efficiency and combustor maximum temperature of Brayton Cycle. It can be seen that on increasing the combustor temperature, efficiency is also increasing. The combined cycle efficiency is 44.08% at combustor temperature 1250K

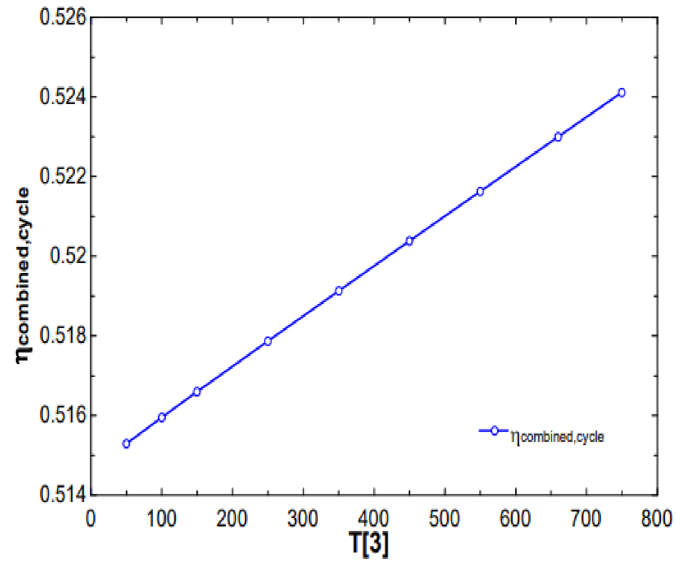

Figure 21: Variation between combined cycle efficiency and turbine inlet temperature of trans-critical Cycle.

Fig. 21 shows the variation between combined cycle efficiency and turbine inlet temperature of trans-critical Cycle. The can be seen that on increasing the temperature, efficiency is also increasing. The combined cycle efficiency is 51.79% at temperature 250ºC.

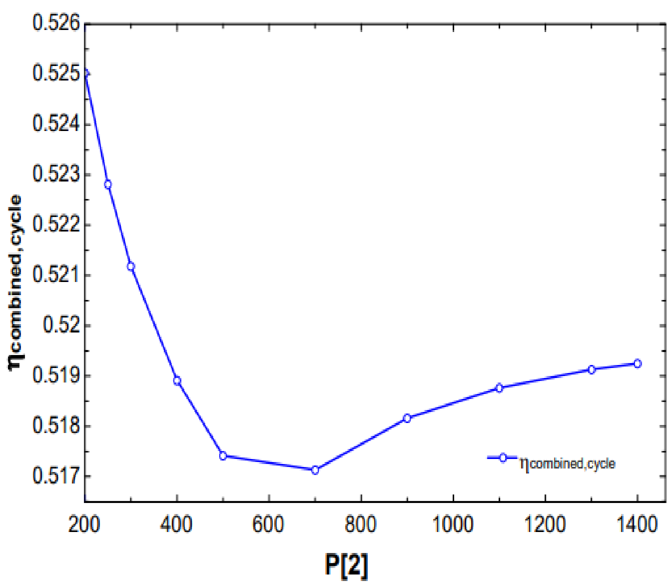

Figure 20: Variation between combined cycle efficiency and exit pressure of pump of trans-critical cycle.

Fig. 20 shows the variation between combined cycle efficiency and exit pressure of pump of trans-critical cycle. It can be seen that on increasing the pressure, efficiency is decreasing and after 700 kPa efficiency is slightly increasing. The combined cycle efficiency is 51.89% at pressure 400 kPa

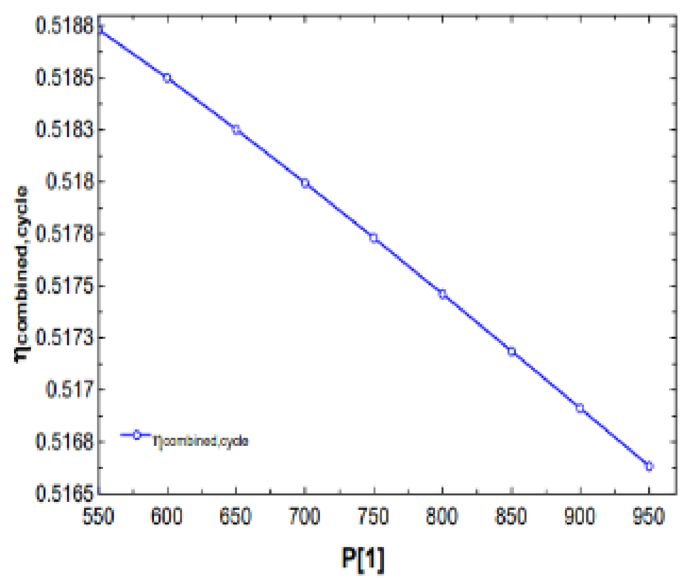

Figure 22: Variation between combined cycle efficiency and inlet pump pressure of trans-critical Cycle

Fig.22 shows the variation between combined cycle efficiency and inlet pump pressure of trans-critical Cycle. It can be seen that on increasing the pressure, efficiency is decreasing. The combined cycle efficiency is 51.8% at pressure 700 kPa.

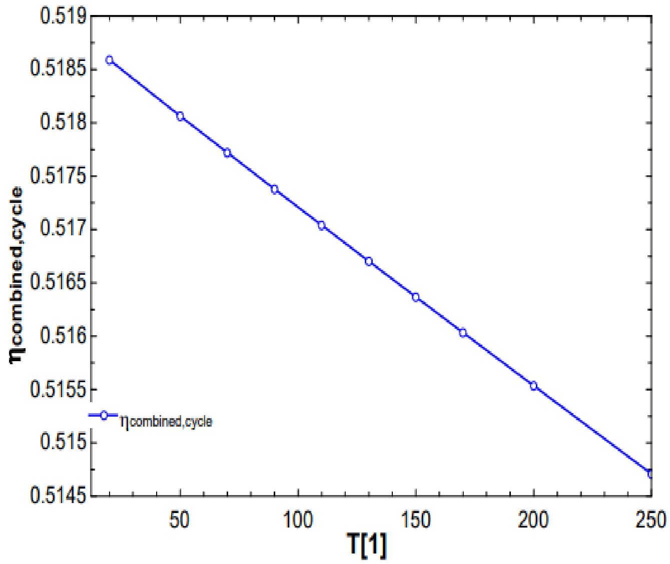

Figure 23: Variation between combined cycle efficiency and turbine inlet temperature of trans-critical Cycle

Fig.23 shows the variation between combined cycle efficiency and turbine inlet temperature of trans-critical Cycle. It can be seen that on increasing the temperature, efficiency is decreasing. The combined cycle efficiency is 51.7% at temperature 110ºC.

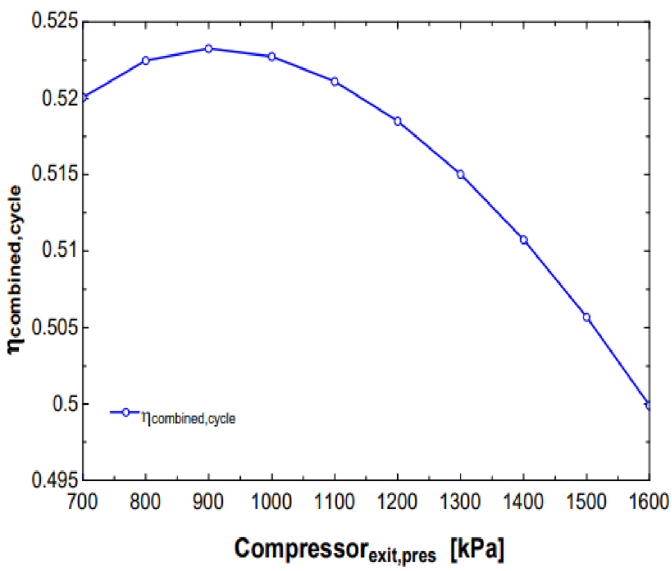

Figure 24: Variation between combined cycle efficiency and compressor exit pressure of Rankine Cycle

Fig. 24 shows the variation between combined cycle efficiency and compressor exit pressure of Rankine Cycle.

It can be seen that on increasing the pressure, efficiency is increasing up to 900 kPa and after that it keeps on decreasing. The combined cycle efficiency is 52.11% at compressor exit pressure 1100 kPa.

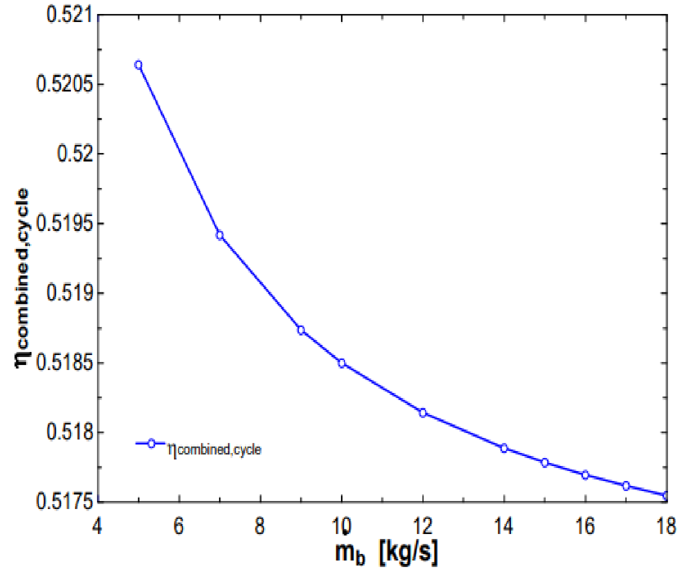

Figure 25: variation between combined cycle efficiency and mass flow rate of Brayton Cycle

Fig. 25 shows the variation between combined cycle efficiency and mass flow rate of Brayton Cycle. It can be seen that on increasing the mass flow rate, efficiency is decreasing. The combined cycle efficiency is 51.81% at mass flow rate 12 kg/sec.

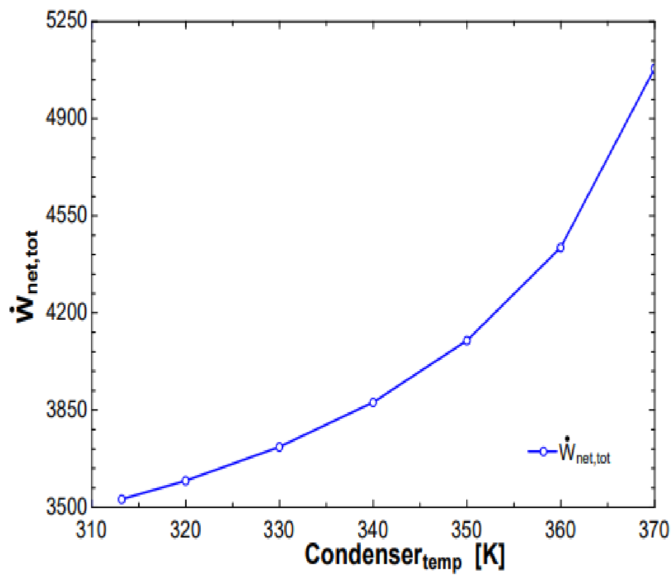

Figure 26: Variation between combined cycle net-work and condenser temperature of Rankine Cycle

Fig.26 shows the variation between combined cycle net-work and condenser temperature of Rankine Cycle. It can be seen that on increasing the condenser temperature, combined net-work is also increasing. The combined cycle net-work is 3878 kW at condenser temperature 340K.

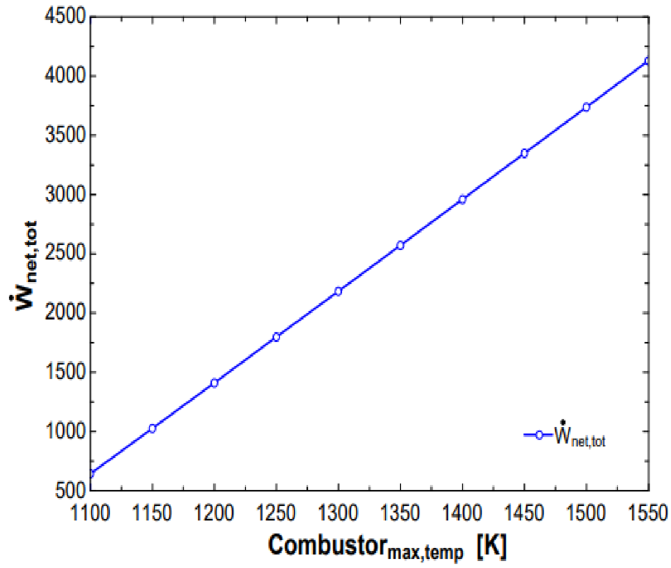

Figure 27: Variation between combined cycle net-work and cycle high temperature of Rankine Cycle

Fig.27 shows the variation between combined cycle net-work and cycle high temperature of Rankine Cycle. It can be seen that on increasing the cycle high temperature, net-work is also increasing. The combined cycle net-work is 3529 kW at temperature 450K.

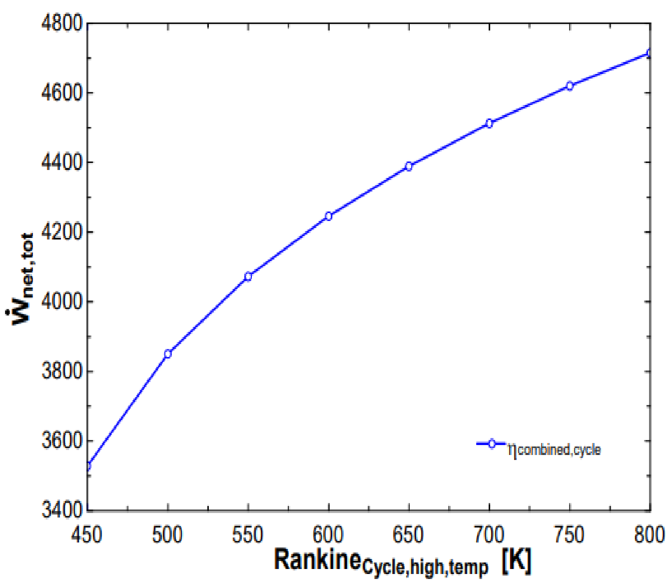

Figure 28: Variation between combined cycle net-work and combustor maximum temperature of Brayton Cycle

Fig. 28 shows the variation between combined cycle net-work and combustor maximum temperature of Brayton Cycle. It can be seen that on increasing the combustor temperature, net-work is also increasing. The combined cycle net-work is 1798 kW at combustor temperature 1250K.

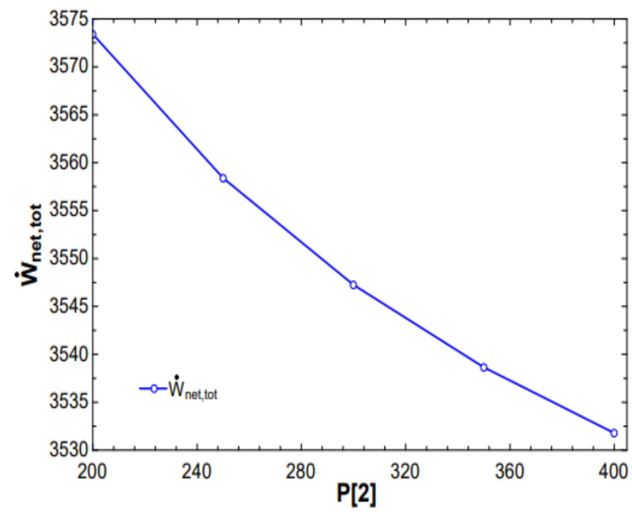

Figure 29: Variation between combined cycle net-work and exit pressure of pump of trans-critical cycle

Fig. 29 shows the variation between combined cycle net-work and exit pressure of pump of trans-critical cycle. It can be seen that on increasing the pressure, net-work is decreasing. The combined cycle net-work is 3532 kW at pressure 400 kPa.

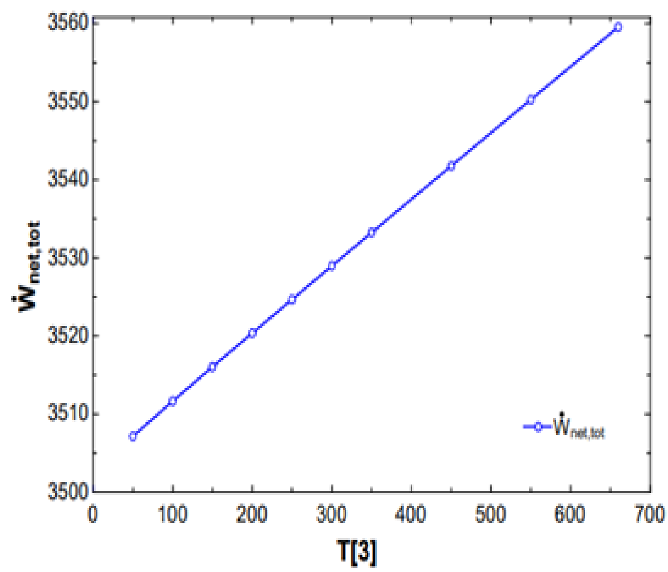

Figure 30: Variation between combined cycle net-work and turbine inlet temperature of trans-critical Cycle

Fig. 30 shows the variation between combined cycle net-work and turbine inlet temperature of trans-critical Cycle. It can be seen that on increasing the temperature, net-work is also increasing. The combined cycle net-work is 3525 kW at temperature 250ºC.

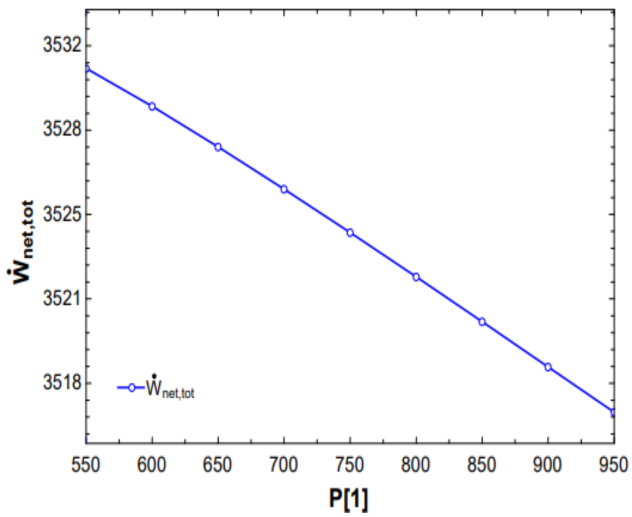

Figure 31: Variation between combined cycle net-work and inlet pump pressure of trans-critical Cycle

Fig.31 shows the variation between combined cycle net-work and inlet pump pressure of trans-critical Cycle. It can be seen that on increasing the pressure, net-work is decreasing. The combined cycle net-work is 3526 kW at pressure 700 kPa

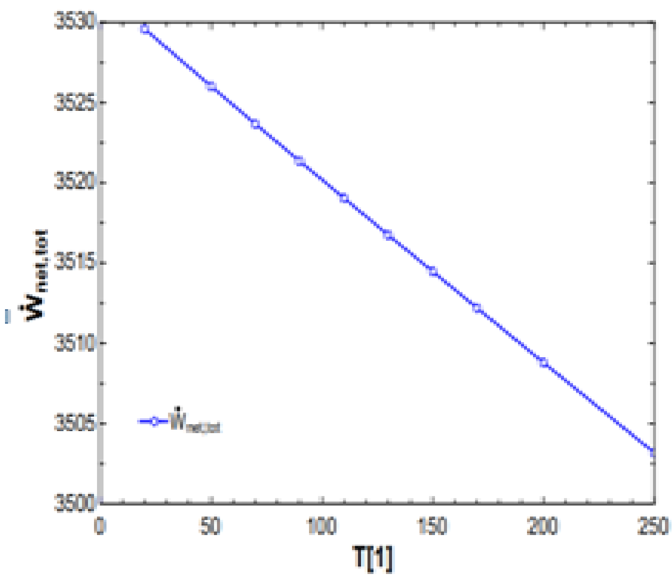

Figure 32: Variation between combined cycle net-work and turbine inlet temperature of trans-critical Cycle

Fig. 32 shows the variation between combined cycle net-work and turbine inlet temperature of trans-critical Cycle. It can be seen that on increasing the temperature, net-work is decreasing. The combined cycle net-work is 3519 kW at temperature 110ºC.

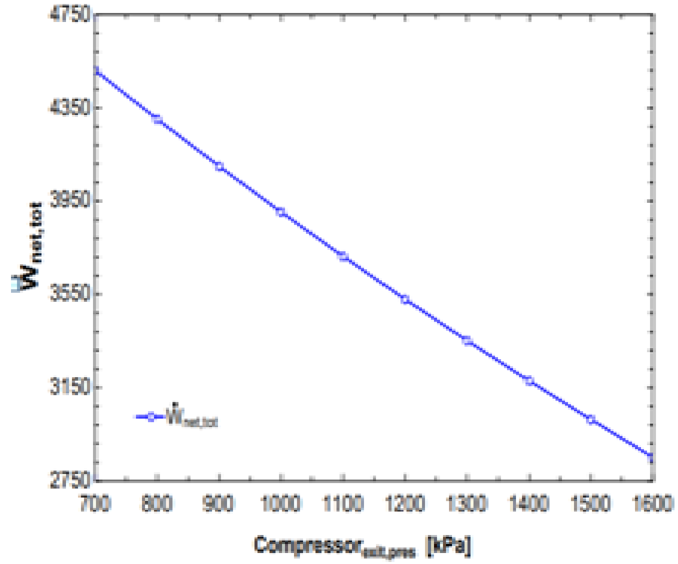

Figure 33: variation between combined cycle net-work and compressor exit pressure of Rankine Cycle

Fig.33 shows the variation between combined cycle net-work and compressor exit pressure of Rankine Cycle. It can be seen that on increasing the pressure, net-work is decreasing. The combined cycle net-work is 3713 kW at compressor exit pressure 1100 kPa.

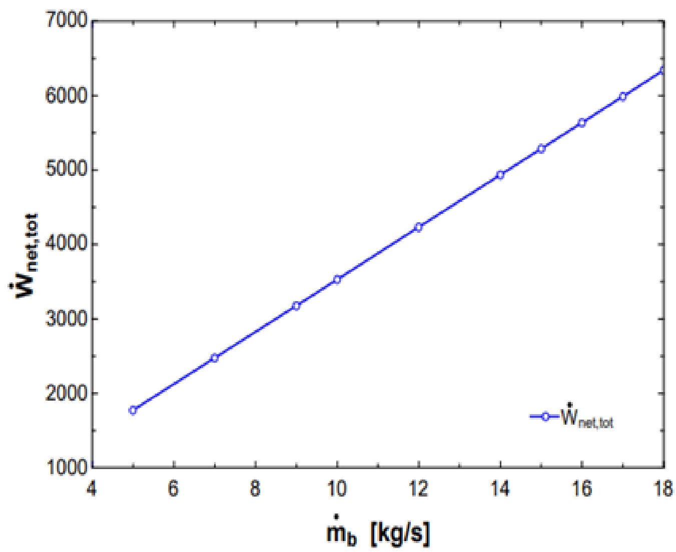

Figure 34: Variation between combined cycle net-work and mass flow rate of Brayton Cycle

Fig. 34 shows the variation between combined cycle net-work and mass flow rate of Brayton Cycle. It can be seen that on increasing the mass flow rate, efficiency is increasing. The combined cycle net-work is 4232 kW at mass flow rate 12 kg/sec

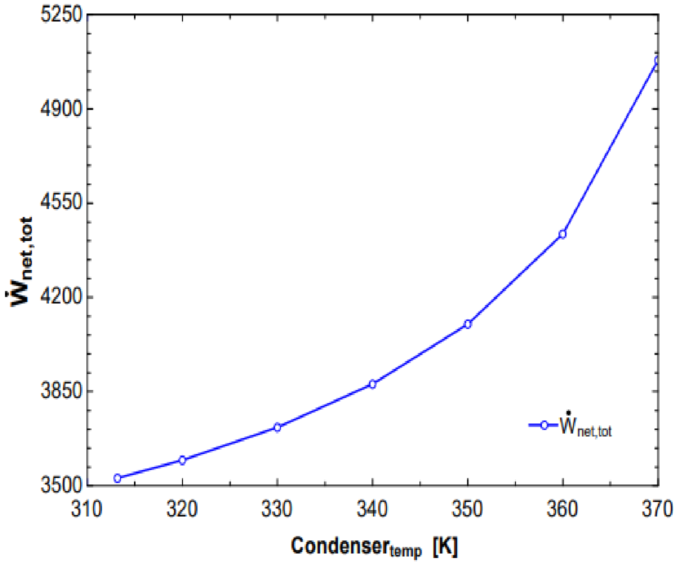

Figure 35: Variation between combined cycle net-work and condenser temperature of Rankine Cycle

Fig.35 shows the variation between combined cycle net-work and condenser temperature of Rankine Cycle. It can be seen that on increasing the condenser temperature, combined net-work is also increasing. From Table 19 combined cycle net-work is 3878 kW at condenser temperature 340K.

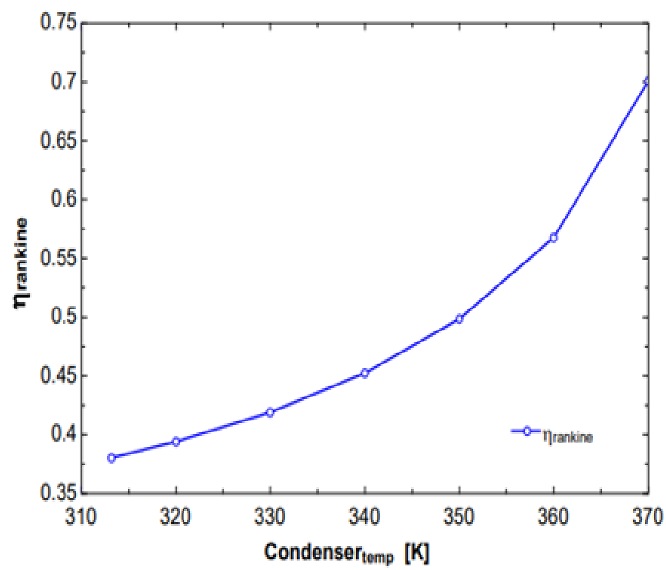

Figure 36: Variation between Rankine cycle efficiency and condenser temperature

Fig.36 shows the variation between Rankine cycle efficiency and condenser temperature. It can be seen that on increasing the condenser temperature, efficiency is also increasing. The Rankine cycle efficiency is 45.25% at condenser temperature 340K

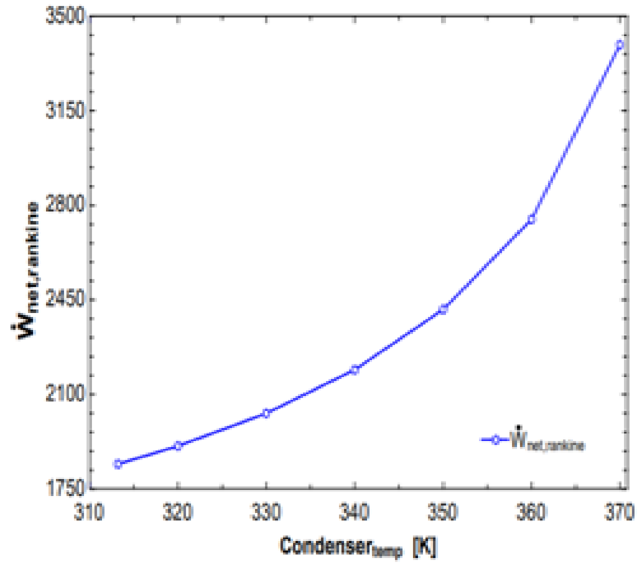

Figure 37: *Variation between Rankine cycle net-work and condenser temperature.*

Fig. 37 shows the variation between Rankine cycle net-work and condenser temperature. It can be seen that on increasing the condenser temperature, Rankine net-work is also increasing. The combined cycle net-work is computed as 2191 kW at condenser temperature 340K.

5. Conclusion

In this study, carbon dioxide cycles is also investigated for recovering the low grade heat. In this thesis, three cycle are combined i.e. Brayton, Rankine and Trans critical cycle in which fluid taken are Ideal gas (air), Steam and Carbon dioxide respectively. The mass flow rate of Brayton cycle is 10 kg/sec and atmospheric temperature is 25ºC.
The following conclusions were drawn from thermodynamic analysis of above mentioned cycles have been shown below.

- On increasing condenser temperature of Rankine cycle, combined efficiency of cycle and combined net-work both are increasing.
- On increasing the mass flow rate of Brayton cycle, combined efficiency of cycle is decreasing and combined net-work is increasing.
- On increasing the exit pressure of pump in trans-critical cycle, combined net work is decreasing also combined cycle efficiency is decreasing.
- On increasing the inlet temperature of turbine of trans-critical cycle, combined cycle efficiency and combined net-work is increasing.
- On increasing the inlet pressure of pump of trans-critical cycle, combined cycle efficiency and combined net work both are increasing.

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Nomenclature

A	Altitude [m]
Ap	Total outer area of the receiver tube [m^2]
ASF	Total solar field aperture area
[m^2]	The specific fuel consumption of the gas turbine unit [tonne/MWh]
BGT	Gas turbine fuel consumption [tonne/h]
C	Solar collector concentration ratio [-]
Cp	Specific heat [kJ/kg.K]
DB	Fuel saving [tonne/h]
Dci	Cover inner diameter [m]
Dco	Cover outer diameter [m]
Devap	Steam mass loss from the deaerator [kg/s]
DFW	Mass flow of feed water [kg/s]
DK	Water mass flow in plant condenser [kg/s]
DLoss	Steam loss [kg/s]
Do	Steam mass flow at the turbine inlet (reference point) [kg/s]
DRK	Water mass flow in GH1 [kg/s]

DRK2 Water mass flow in GH2 [kg/s]
DRT The extracted steam to operate the plant deaerator [kg/s]
DSS The generated stem due to solar field Contribution [kg/s]
Dti Receiver inner diameter [m]
Dto Receiver outer diameter [m]
h' Saturated water specific enthalpy [kJ/kg]
h" Saturated steam specific enthalpy [kJ/kg]
Ib Beam solar radiation [W/m^2]
Id Diffuse solar radiation [W/m^2]
Isc Solar constant [W/m^2]
Iso Extra-terrestrial solar radiation [W/m^2]
K Receiver thermal conductivity [W/m. °K]
kc Cover thermal conductivity [W/m. °K]
Ke Cover extinction coefficient [m-1]
l Collector length [m]
N Number of collectors in each row [-].
M Water mass flow for each row in the solar field [kg/s] m.
SF Water mass flow for whole solar field [kg/s]
mC Relative air mass flow for blades cooling in gas turbine unit
mf The relative fuel mass flow for gas turbine unit [kg fuel/kg air]
mgas Gases mass exhaust from gas turbine unit [kg/s] mK
N Number of rows of solar field [-]
n2 Cover refractive index [-]
NEGT Gas turbine output [MW]
NEST Steam turbine output [MW]
NFP Energy consumption by water feed pump [MW]
P Pressure [bar]
PD Deaerator pressure [bar]
PDE Deaerator's evaporator pressure [bar]
Pk Condenser pressure [bar]
PLPT Pressure at LPT inlet [bar]
PLPTO Pressure at LPT inlet for combined cycle operation [bar]
PSOSF Design outlet pressure for solar field [bar]
Qc.v Fuel calorific value [kJ/kg]
QL Heat loss from solar collector [kW]
QSC Useful heat from solar field [kW]
QSF Nominal solar field output [kW]
Qu Useful heat gain in solar field (for each row) [kW]
R Gas constant [kJ/kg.°K]
S Specific entropy [kJ/kg.°K]
Sb Absorbed solar energy by receiver tube [W/m^2.°K]
T Temperature [°C]
Ta Ambient temperature [°C]
Tbw The average temperature of gas turbine blades [°C]
Tex Exhaust Gases temperature after HRSG °C]
Tfi Water temperature at solar field inlet [°C]
Tfo Water temperature at solar field outlet [°C]
TL Disposed water temperature [°C]
TRFW1 Re-feed water temperature [°C]
TS Temperature for ideal process (isentropic) [°C]
Ua Wind Velocity [m/s]
UL Solar collector loss coefficient [W/m^2.°K]
W Collector aperture width [m]
Wa Specific work done by gases in the GT [kJ/kg]
Wco Specific work done by cooling air in the GT [kJ/kg]
We Specific work for gas turbine unit [kJ/kg]
WK Compressor specific work [kJ/kg]
WT Total specific work of GT (gases + air) [kJ/kg]
XK Steam/water dryness factor [%]

Abbreviations

AC Air compressor
ANU The Australian National University
CC Combined cycle
CHP Combined Heat and Power
CSES Centre for Solar Energy Studies – Libya
CSP Concentrating Solar Power
DE The evaporator of the desecrator
DISS Direct solar steam European project
DLR German Aerospace Centre
DSG Direct steam Generation
EC European commission
ECC Equivalent combined cycle
ETB Engineering tool Book
EU-MENA Europe, Mediterranean North African region
FP Feed water pump
FV Flash vessel
G Electricity generator
GCC Gas Turbine Combustion chamber
GECOL General Electricity Company of Libya
GH Gas heater
GT Gas turbine
GTU Gas turbine unit
HPT High pressure turbine
HRSG Heat Recovery Steam Generation
HTF Heat Transfer Fluid
HVDC High Voltage Direct Current
ISCC Integrated Solar Combined Cycle Power Plant
LPT Low pressure turbine
LREC Libyan Renewable Energy Centre
MED Multi Effect Desalination Unit
MSF Multi Stage Flash Desalination Unit
NREL National Renewable Energy Laboratory
RFWH Re-feed water heater
SEEN The Sustainable Energy and Economy Network
SEGS Solar Electricity Generating Station
STU Steam turbine unit
SV Separator vessel
TRANS-CSP Trans-Mediterranean Interconnection for Concentrating Solar Power
TREC Trans-Mediterranean Renewable Energy Cooperation