"Thermodynamic analysis and comparison of various organic fluids for ORC in Gas turbine-Organic Rankine combined cycle plant with solar reheating and regeneration of ORC fluid"

Major project-II

Submitted to Delhi Technological University in partial fulfillment of the requirement for the award of Degree of

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UNDER THE SUPERVISION OF

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CANDIDATE'S DECLARATION

I hereby declare that the work presented in this thesis titled "**Thermodynamic analysis and comparison of various organic fluids for ORC in Gas turbine-Organic Rankine combined cycle plant with solar reheating and regeneration of ORC fluid**", in the partial fulfillment for the award of the degree of Master of Technology in "THERMAL ENGINEERING" submitted to Department of Mechanical Engineering, is an authentic record of my own work carried out under the supervision of prof. Dr. R. S. MISHRA, Department of Mechanical Engineering, Delhi technological university, Delhi. This report does not, to the best of my knowledge, contain part of my work which has been submitted for the award of any other degree either of this university or any other university without proper citation.

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This is to certify that above statement made by "DHARMENDRA SAHU" is true to the best of my knowledge and belief.

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ABSTRACT

The integration of solar energy into natural gas combined cycles has been successfully demonstrated in several integrated solar combined cycles since the beginning of this decade in many countries. There are many motivations that drive investments on integrated solar combined cycles which are primarily the repowering of existing power plants, the compliance with more severe environmental laws on emissions and the mitigation of risks associated with large solar projects. Integrated solar combined cycles are usually developed as brown field facilities by retrofitting existing natural gas combined cycles and keeping the existing equipment to minimize costs. In this work energy and exergy analysis of Gas turbine-Organic Rankine combined cycle with solar reheating of organic fluid is done and results are compared with simple combined cycle, combined cycle with regeneration, combined cycle with solar reheating of organic fluid. The performance of the system is compared with different organic fluids such as R134a, R245fa, Acetone, and R1234yf at different organic Rankine cycle maximum pressure and maximum temperature. R1234yf shows maximum increase in efficiency by regeneration about 70%. Acetone shows maximum organic Rankine cycle efficiency of 25.96%. exergetic efficiency of combined cycle with regeneration and reheating 64%, 72%, 64%and 82% for R134a, R245fa, R1234yf and Acetone respectively. Acetone is recommended for practical applications due to its highest energy and exergetic efficiency among all selected organic fluids but some important problem related to flammability and explosion risk have to be considered while managing it. After Acetone, R245fa can be considered as better option in solar reheated combined cycle plant with regeneration.

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NOMENCLATURE

h	Specific Enthalpy (kJ/kg)
s	Specific Entropy (kJ/kgK)
m	Mass flow rate (kg/s)
P	Pressure (kPa)
T	Temperature (^{0}C)
Q	Heat Exchange per unit mass (kW)
W	Work (kW)
η	Isentropic Efficiency(dimensionless)

Subscript/Superscript

C	Compressor
ORC	Organic Rankine cycle
GTP	Gas Turbine Plant
CCP	Combined Cycle Plant
HRB	Heat Recovery Boiler
GT	Gas Turbine
OT	Organic Turbine
Cond	Condenser
reg	regenerator
f	Organic fluid
a	Air
ET	Exergy Transfer
OP	Organic Pump

CHAPTER 1

INTRODUCTION

Several integrated solar combined cycles (ISCCs) are in operation all around the world (North Africa, Iran, Italy, USA) and many projects are underway (Mexico, China, USA). ISCCs offer many advantages compared to solar thermal power plants, primarily associated with the higher solar energy conversion efficiency and the lower investment costs. Investors and owners are attracted by the mitigated risk associated with the construction of smaller solar fields compared to solar thermal power plants.

The process for converting the energy in a fuel into electric power involves the creation of mechanical work, which is then transformed into electric power by a generator. Depending on the fuel type and thermodynamic process, the overall efficiency of this conversion can be as low as 30 percent. This means that two-thirds of the latent energy of the fuel ends up wasted. For example, steam electric power plants which utilize boilers to combust a fossil fuel average 33 percent efficiency. Simple cycle gas turbine (GTs) plants average just under 30 percent efficiency on natural gas, and around 25 percent on fuel oil. Much of this wasted energy ends up as thermal energy in the hot exhaust gases from the combustion process.

To increase the overall efficiency of electric power plants, multiple processes can be combined to recover and utilize the residual heat energy in hot exhaust gases. In combined cycle mode, power plants can achieve electrical efficiencies up to 60 percent. The term “combined cycle” refers to the combining of multiple thermodynamic cycles to generate power. Combined cycle operation employs a heat recovery boiler (HRB) that captures heat from high temperature exhaust gases to produce organic vapour, which is then supplied to a organic turbine to generate additional electric power. The process for creating vapour to produce work using a organic turbine is based on the Rankine cycle..

The most common type of combined cycle power plant utilizes gas turbines and is called a combined cycle gas turbine (CCGT) plant. Because gas turbines have low efficiency in simple cycle operation, the output produced by the steam turbine accounts for about half of the CCGT plant output. There are many different configurations for CCGT power plants, but typically each GT has its own associated HRB, and multiple HRBs supply vapour to one or more organic turbines.

1.1 Organic Rankine Cycle:-

Organic Rankine cycles have received much attention during last decade. This cycle obeys the fundamental rules of conventional Rankine cycles working with water in common plants but has some advantages over water Rankine cycle which made it popular. First this cycle can work under low temperatures and pressures in comparison to conventional Rankine cycle and shows a better performance than water especially from low grade heat sources because its working fluids include a variety of hydrocarbons and refrigerants and according to the range of accessible heat source temperatures and pressures, different outputs can be derived by using suitable working fluids, second, it can work without feed-water heaters and multi-stage turbines which makes it simple to use. Among these, solar parabolic trough collectors are a huge source of thermal energy but with a low grade heat which makes it only suitable for some kilowatts to few megawatts electricity generation, also it shows a reliable means for electricity generation especially in rural areas or near factories to generate their electricity consumption without the need for connection to grid which may be expensive. Disadvantages of solar ORCs are relatively high costs and low thermal efficiency (10 to 25 percent according to working fluids and working condition) mainly because of low HTF (Heat transfer fluid) temperature in solar collectors. As mentioned before, the organic fluids used in ORC cycles are divided into hydrocarbons and refrigerants, some of them are dry fluids which mean they have a positive slope T-S diagram in the saturation vapour region. This makes it possible for some organic fluids to work properly without superheating to a great extent and cause no damage to turbine. As shown in this study, a comparison of different dry organic fluids with and without superheating and recuperation has been done to show the variance in cycle efficiency and performance of the system which helps us to make a decision to choose the system conditions according to our needs.

1.2 Combined Cycle Principles of Operation:-

The organic Rankine cycle's principle is based on a turbogenerator working as a normal steam turbine to transform thermal energy into mechanical energy and finally into electrical energy through an electrical generator. Instead of using water steam, organic Rankine cycle system vaporizes organic fluid, characterized by a molecular mass higher than water, which leads to slower rotation of the turbine and lower pressure and erosion of the metallic parts and blades. Organic Rankine cycle is simply a Rankine cycle in which instead of using water other organic fluid is used such as R134a, Acetone, R1234a, R245fa etc.

The HRB is basically a heat exchanger, or rather a series of heat exchangers. It is also called a boiler, as it creates organic vapour for the organic turbine by passing the hot exhaust gas flow from a gas turbine or combustion engine through banks of heat exchanger tubes. The HRB can rely on natural circulation or utilize forced circulation using pumps. As the hot exhaust gases flow past the heat exchanger tubes in which hot water circulates, heat is absorbed causing the creation of vapour in the tubes. The tubes are arranged in sections, or modules, each serving a different function in the production of dry superheated vapour. These modules are referred to as economizers, evaporators, superheater and preheaters.

The economizer is a heat exchanger that preheats the organic fluid (liquid) to approach the saturation temperature (boiling point), which is supplied to a thick-walled boiler drum. The drum is located adjacent to finned evaporator tubes that circulate heated organic fluid. As the hot exhaust gases flow past the evaporator tubes, heat is absorbed causing the creation of vapour in the tubes. The vapour-liquid mixture in the tubes enters the boiler drum where vapour is separated from the hot liquid using moisture separators and cyclones. The separated liquid is recirculated to the evaporator tubes. Boiler drums also serve storage and water treatment functions. An alternative design to steam drums is a once-through HRB, which

replace the boiler drum with thin-walled components that are better suited to handle changes in exhaust gas temperatures and vapour pressures during frequent starts and stops. In some designs, duct burners are used to add heat to the exhaust gas stream and boost vapour production; they can be used to produce vapour even if there is insufficient exhaust gas flow.

Saturated vapour from the boiler drums or once-through system is sent to the superheater to produce dry vapour which is required for the organic turbine. Preheaters are located at the coolest end of the HRB gas path and absorb energy to preheat heat exchanger liquids, such as water/glycol mixtures, thus extracting the most economically viable amount of heat from exhaust gases.

The superheated organic vapour produced by the HRB is supply to the organic turbine where it expands through the turbine blades, imparting rotation to the turbine shaft. The energy delivered to the generator drive shaft is converted into electricity. After exiting the organic turbine, the organic vapour is sent to a condenser which routes the condensed organic liquid back to the HRB

Combined-cycle plant works to produce electricity and captures waste heat from the gas turbine to increase efficiency and electrical output.

1. Gas turbine burns fuel.

   - The gas turbine compresses air and mixes it with fuel that is heated to a very high temperature. The hot air-fuel mixture moves through the gas turbine blades, making them spin.
   - The fast-spinning turbine drives a generator that converts a portion of the spinning energy into electricity.

2. Heat recovery system captures exhaust.

   - A Heat Recovery Boiler (HRB) captures exhaust heat from the gas turbine that would otherwise escape through the exhaust stack.
   - The HRB creates organic vapour of organic fluid from the gas turbine exhaust heat and delivers it to the steam turbine.

3. Organic turbine delivers additional electricity.

   - The organic turbine sends its energy to the generator drive shaft, where it is converted into additional electricity.

1.3 Combined cycle plant without regeneration:

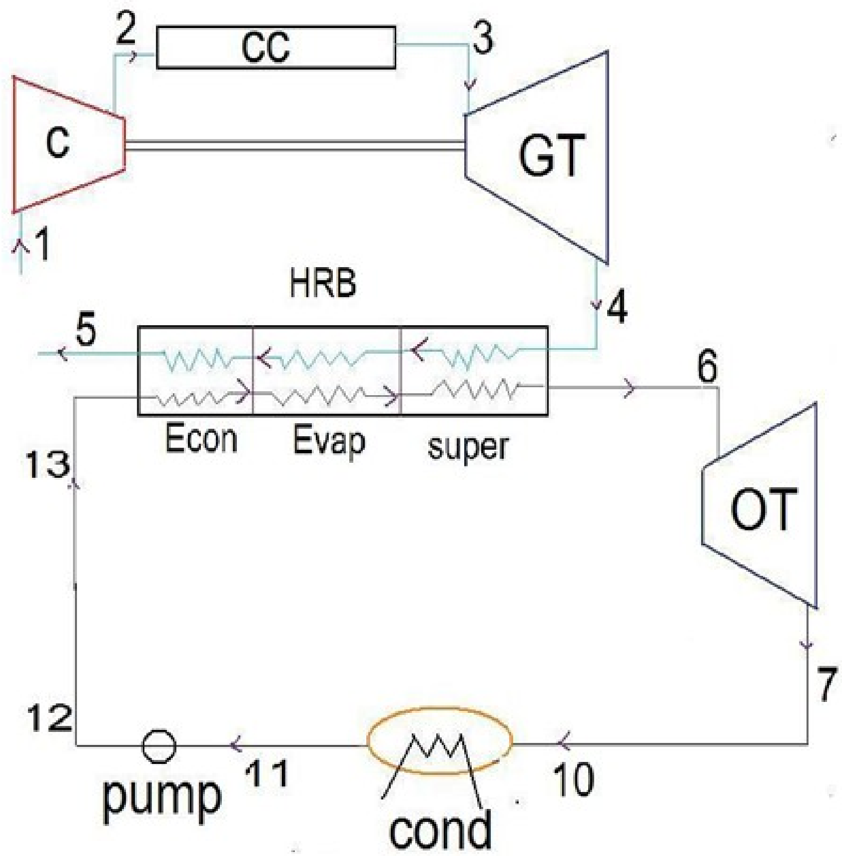

Fig 1: combined cycle plant without regeneration

Combined cycle power plants, exhaust gas from GT are at considerably high temperature. This exhaust gas heat is utilized in HRB to boil organic fluid in ORC. Temperature of condenser is always taken to be 40˚C for proper heat transfer. Pressure drop in various ducts are neglected.

1.4 Combined cycle plant with regeneration:

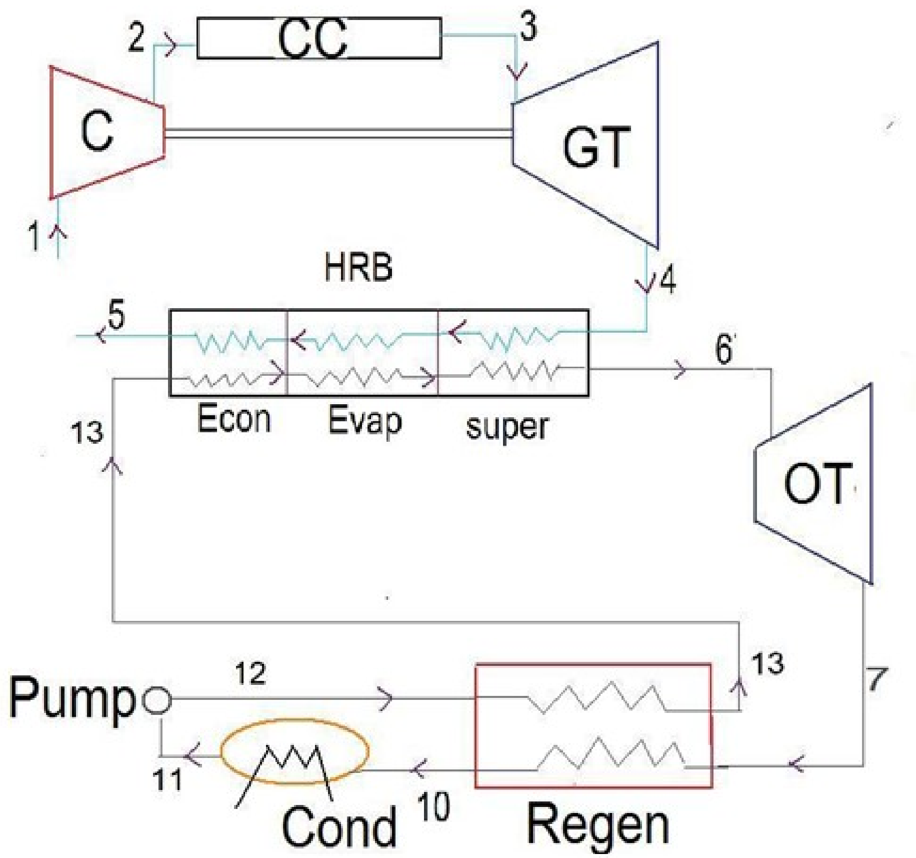

Fig 2: combined cycle plant with regeneration

A regenerator has been installed which heat the organic fluid (liquid) by the high temperature vapour leaving the organic turbine. In some organic fluid when expanded in organic turbine, at exit give superheated vapour this vapour enthalpy is utilized in heating feed organic fluid. Which results the heat addition required in boiler is reduced. The condenser temperature is again set to 40˚C.

1.5 Combined cycle power plant with solar reheating and regeneration:-

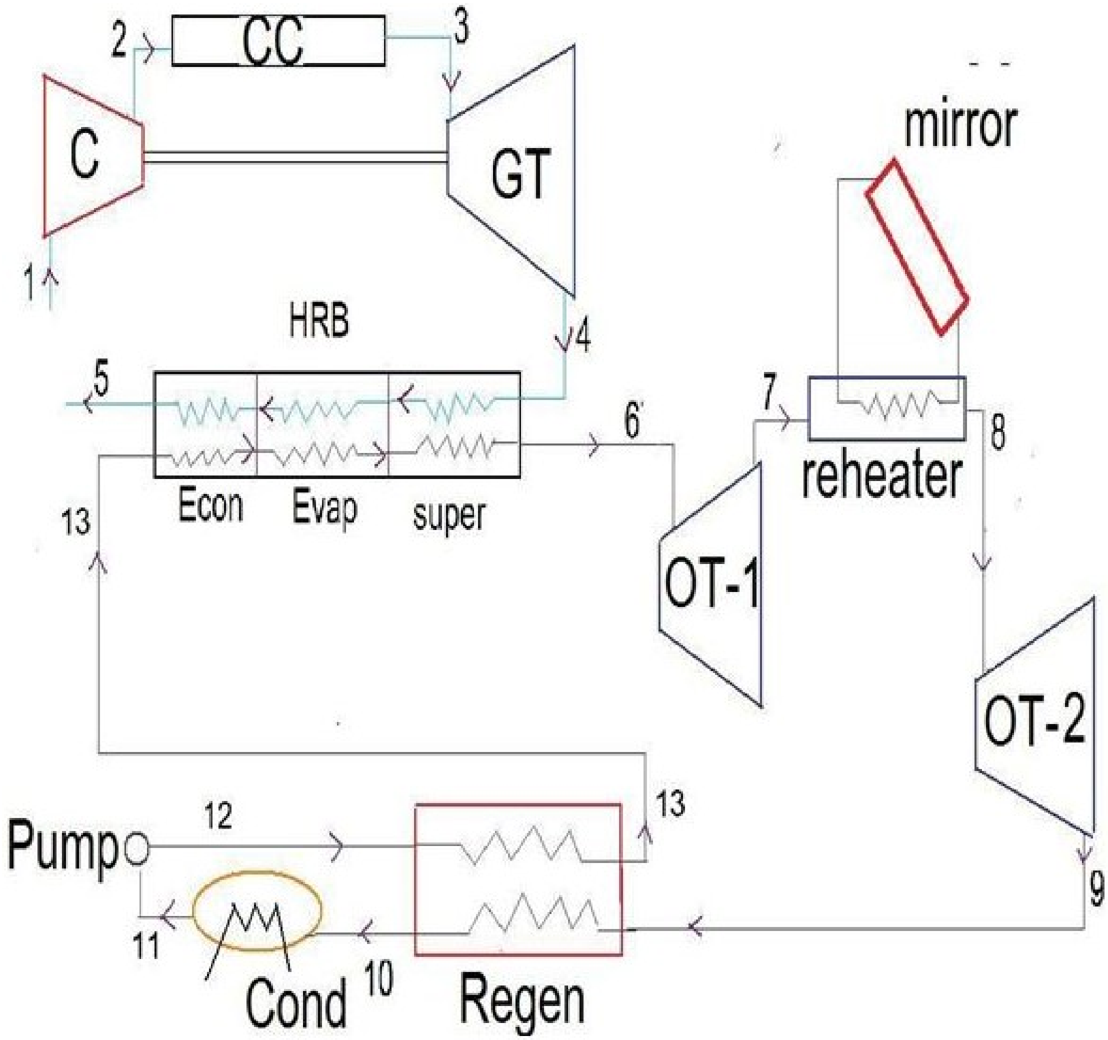

Fig3: combined cycle plant with regeneration and solar reheating

In this cycle solar energy is used for reheating the organic fluid in dual pressure organic Rankine cycle plant. A concentrated solar plate collector is used for capturing solar energy. The condenser temperature is taken to be 40˚C in each case in order to maintain heat transfer from condenser to environment. It is assumed that total pressure drop in organic Rankine cycle are equally divided in high pressure and low pressure organic Rankine turbines.

Utilization of solar energy has become crucial and it is expected to increase significantly in the near future. Therefore, there is a need to improve the performance of thermal power plants integrated with solar thermal energy. Parabolic trough solar collector technology is considered the most established solar thermal technology for power production. It has been used in large power plants since the 1980s in California and has demonstrated a promising renewable energy technology for the future.

CHAPTER 2

LITERATURE REVIEW

In this chapter, it is intended to give a brief literature review of work being carried out on solar integrated Gas turbine-Organic Rankine cycle power plant. Many papers have appeared since the late nineties about the thermodynamic analysis of ISCCs focusing on integration of solar energy into the combined cycle.

2.1 LITERATURE

Kelly et al [1] 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 levelized cost of solar energy.

He Ya Ling et al [2] 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 (q_{loss}) increases sharply with the increase in p_{inter} at beginning and then reaches to an approximately constant value. The variation of heat collecting efficiency (η_{hc}) with v is quite similar to the variation of q_{loss} with p_{inter}. However, I_{dn} and θ exhibit opposite effect on η_{hc}. In addition, it is found that the optimal volume of the thermal storage system is sensitively dependent on the solar radiation intensity. The optimal volumes are 100, 150, 50, and 0 $m^3$for spring equinox, summer solstice, autumnal equinox and winter solstice, respectively.

Gang et al. [3] proposed 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 the regenerative 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/m^2 and is relatively higher than that without the regenerative cycle by 4.9%.

Manolakos D et al [4] proposed co-generation system producing electricity and freshwater 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 hexamethyldisiloxane (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.

Nafey and Sharaf [5] carried out design and performance analysis using *MatLab/SimuLink* computational environment. The cycle consists of thermal solar collectors (Flat Plate Solar Collector (FPC), or Parabolic Trough Collector (PTC), or Compound Parabolic Concentrator (CPC)) for heat input, expansion turbine for work output, condenser unit for heat rejection, pump unit, and Reverse Osmosis (RO) unit. Reverse osmosis unit specifications used in this work is based on Sharm El-Shiekh RO desalination plant. Different working fluids such as: butane, isobutane, propane, R134a, R152a, R245ca, and R245fa are examined for FPC. R113, R123, hexane, and pentane are investigated for CPC. Dodecane, nonane, octane, and toluene are allocated for PTC. Exergy and cost analysis are performed for saturation and superheated operating conditions. Toluene and Water achieved minimum results for total solar collector area, specific total cost and the rate of exergy destruction.

Sharaf et al [6] carried out thermo-economic analysis of PTSC integrated with an ORC and a multi-effect distillation. Two scenarios of generation were considered in their study: the first one was with only water production and the second one was with both power and water production. The comparison is implemented according to the operation of Parabolic Trough Collector (PTC) with toluene organic oil and water working fluids. Therminol-VP1 Heat Transfer Oil (HTO) is considered for indirect vapor generation operation across the solar field and evaporator heat exchanger. The comparisons are manipulated according to

100 m^3/day of distillate product as a case study. As a result, only desalination technique is considered more attractive than desalination and power technique due to higher gain ratio and lower solar field area needed.

Delgado-Torres and Garcia Rodriguez [7] & [8] conducted thermodynamic analysis of a thermal system which consists of an ORC, a PTSC, and an RO (Reverse Osmosis). Initially they analyzed the system assuming only water production through RO then they extended their study to include both electrical and water production the main objective of their study was to examine the effect of different organic fluids on the aperture area of the PTSC.

Al-Sulaiman et al. **[9] & [10]** proposed the energetic performance analysis of PTSC integrated with an ORC in which the waste heat from the ORC is used for cogeneration was conducted. It was found that there was an energy efficiency improvement, when trigeneration was used, from 15% to 94% (utilization efficiency). On the other hand using exergy analysis, found that there was an exergetic efficiency improvement from 8% to 20% when trigeneration is used as compared to only power generation.

Al-Sulaiman [11] carried out solar field sizing and overall performance analysis of different vapor cycles. The systems considered are parabolic trough solar collectors integrated with either a binary vapor cycle or a steam Rankine cycle (SRC). The binary vapor cycle consists of an SRC as a topping cycle and an organic Rankine cycle as a bottoming cycle. Seven refrigerants are examined for the bottoming cycle: R600, R600a, R134a, R152a, R290, R407c, and ammonia. Founds that significant reduction in the solar field size is gained due to the performance improvement when the binary vapor cycle is considered as compared to a steam Rankine cycle with atmospheric condensing pressure; however, SRC with vacuum pressure has the best performance and smallest solar field size. It further reveals that the R134a binary vapor cycle has the best performance among the binary vapor cycles considered and, thus, requires the smallest solar field size while the R600a binary vapor cycle has the lowest performance.

Fahad A. Al-Sulaiman [12] carried out detailed exergy analysis of thermal power system driven by parabolic trough solar collectors (PTSCs). The power is produced using either a steam Rankine cycle (SRC) or a combined cycle, in which the SRC is the topping cycle and an organic Rankine cycle (ORC) is the bottoming cycle. Seven refrigerants for the ORC were examined: R134a, R152a, R290, R407c, R600, R600a, and ammonia. The R134a combined cycle demonstrates the best exergetic performance with a maximum exergetic efficiency of 26% followed by the R152a combined cycle with an exergetic efficiency of 25%. Alternatively, the R600a combined cycle has the lowest exergetic efficiency, 20–21%.

Milad Ashouri et al [13] carried out analysis of a photovoltaic through collector (PTC) integrated with an organic Rankine cycle (ORC) for small scale electricity generation near Tehran. The system includes a solar field, a storage tank, and a small scale ORC engine. Performance evaluation has been done by means of commercial software Thermoflex19. A comparison of different working fluids is presented and results shows that Benzene has the best performance among fluids butane, n-pentane, isopentane, R123 and R245fa for the system conditions described.

Dimityr Popov [14] proposed a concept for innovative hybridization of gas turbine combined cycle plant and solar power system. This conceptual plant is named as Solar Assisted Combined Cycle, as the solar energy is indirectly involved in power generation. The proposed solar hybridization can be accomplished in two ways. The first solar assisted option introduces mechanical chillers for a complete cooling of gas turbine inlet air. The next solar assisted option does the same but with an absorption chiller. They find that the configuration with absorption chillers has lower specific incremental plant capital costs and requires smaller land area than the others.

Wang et al [15] analyzed a 1.6 kWe solar ORC using a rolling piston expander. An overall efficiency of 4.2% was obtained with evacuated tube collectors and 3.2% with flat-plate collectors. The difference in terms of efficiency was explained by lower collector efficiency (71% for the evacuated tube vs. 55% for the plate technology) and lower collection temperature.

S. Quoilin et al [16] carried out thermodynamic modeling of a proposed small scale PTSC integrated with an ORC for power production, considering different design options to be located in a rural location in South Africa and presented an optimization and sizing procedure of heat exchangers in a small scale solar driven ORC by pinch and pressure drop and optimized it by turbine input pressure and evaporator temperature. Then they further focused on the evaluation of the thermodynamic performance of the system. With conservative hypotheses, and real expander efficiency curves, it was found that an overall electrical efficiency between 7% and 8% reached. This efficiency is steady-state efficiency at a nominal working point. The comparison between working fluids showed that the most efficient fluid is Solkatherm. R245fa also shows a good efficiency and has the advantage of requiring much smaller equipment.

2.2 CONCLUSION AND GAP

Combined cycle power plant is used for producing power by coupling Gas turbine cycle (Top cycle) and Organic Rankine cycle (bottom cycle). By incorporation of solar energy in combined cycle plant the output of plant can be increased. From literature survey of solar integrated combined cycle plant, it is found that detailed energy and exergy analysis of Gas-turbine power plant, Organic Rankine cycle plant and combined cycle power plant are done earlier. From literature survey it is found that energy and exergy analysis of solar reheated organic Rankine cycle combined with Gas-turbine cycle are not done yet. Solar energy is used for reheating the organic vapour leaving the high pressure organic turbine and after reheating supplied to low pressure organic turbine.

2.3 PROBLEM FORMULATION

In the present study thermodynamic analysis of Gas turbine-Organic Rankine combined cycle plant with solar reheating is investigated and comparison of various organic fluids is carried out to find best organic fluid which will give maximum efficiency. The effect of regeneration and reheating are also evaluated on performance of combined cycle plant.

It is proposed to examine the effect of following parameters on the efficiency of combined cycle and Organic Rankine cycle plant of solar reheated combined cycle plant and evaluation of better organic fluid.

- Effect of using regenerator and solar reheater in system on efficiency and exergy destruction.
- Effect of Organic turbine inlet temperature.
- Effect of Organic turbine inlet pressure.
- Effect of various organic fluids.

CHAPTER-3

SYSTEM DESCRIPTION

For thermodynamic analysis of combined cycle plant an existing 25MW Gas turbine power plant is selected. For this toping cycle an Organic Rankine cycle is combined with HRB and the effect of regeneration, solar reheating, and various organic fluids is evaluated. Efficiency of Organic Rankine cycle and combined cycle plant are analyses for different boiler pressure superheated cycles of with and without regeneration and solar reheating. The following organic fluid R134a, R1234yf, R245fa, Acetone is analyzed in this system.

Gas turbine-Organic Rankine combined cycle plant shown in below figure. Air from atmospheric pressure and temperature is compressed adiabatically in air compressor from1 to 2 and consumes Wc. this compressed air is heated at constant pressure in external heat exchanger from 2 to 3 where the temperature of air is increased and Q_{s1} heat supplied. Here composition of air remains same no combustion product mixed with compressed air, this heated air is expanded in Gas turbine from 3 to 4 and work is obtained W_{GT}. Some work is used in driving the compressor.

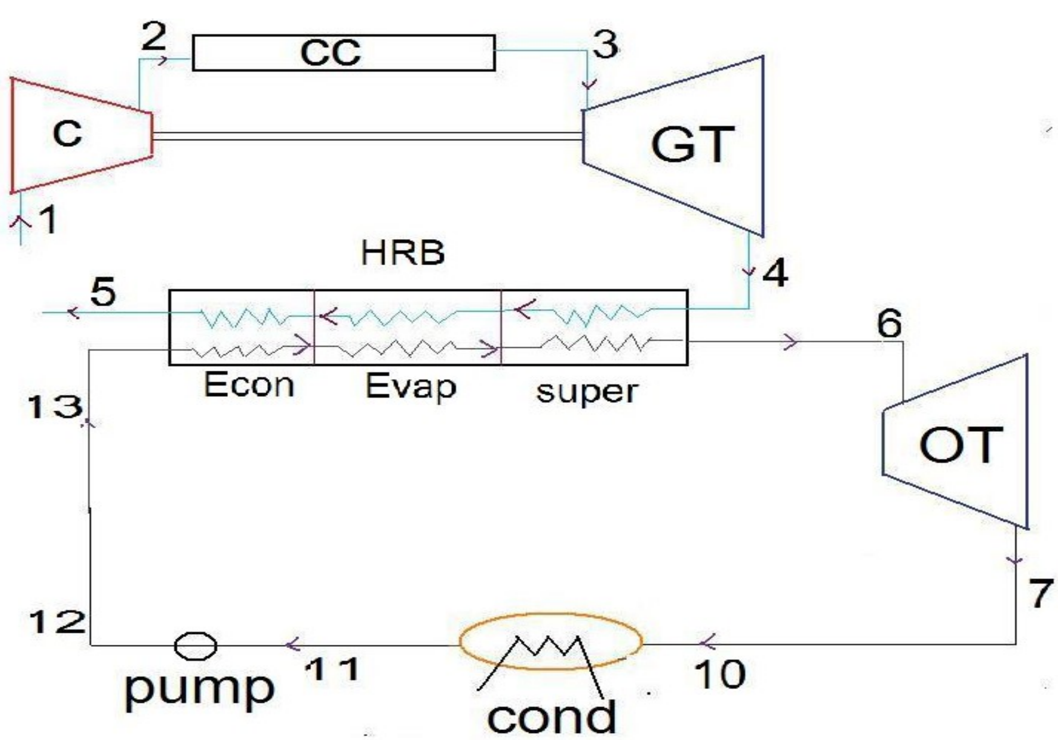

Fig4: combined cycle plant without regeneration

Exhaust gas from Gas turbine have considerably high temperature and energy. This energy is used in Organic Rankine cycle for boiling the organic fluid. This energy is utilised in HRB to heat Organic fluid to produce superheated vapour by absorbing Q_{s2} heat from 4 to 5. This superheated vapour is expanded in organic turbine from 6 to 7 and W_{OT} work is extracted. There is condenser at the exit of organic turbine where organic fluid is condensed from 10 to 11 to get saturated organic liquid. Than this liquid is pumped from condenser pressure to boiler pressure in pump from 11 to 12. There after organic fluid is heated in HRB by hot exhaust gas of Gas turbine plant and superheated vapour is produced from 13 to 6, and supplied to organic turbine.

In steam Rankine cycle steam at the exit of steam turbine is in wet region generally while in organic Rankine cycle, organic fluid is used which is in superheated region at exit of organic turbine at state 10. This enthalpy of superheated vapour is utilized in heating the feed liquid in liquid vapour heat exchanger also called regenerator. In regenerator Organic liquid is heated from state 12 to state 13 and Organic vapour cools from state 7 (or 9) to state 10 at constant pressure. By using regenerator in combined cycle plant rate of organic fluid vapour production increases which results work output increased.

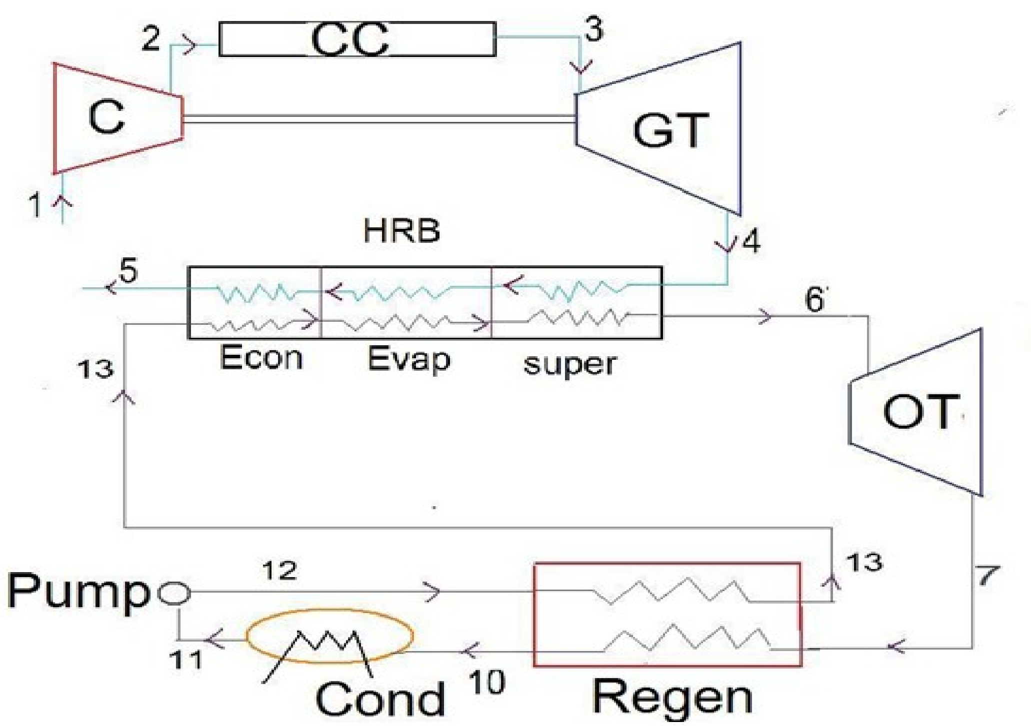

Fig5: combined cycle plant with regeneration

As we know that solar energy is most abundantly available, if this energy is utilized in combined cycle plant than output of the plant is increased. In this project dual pressure organic Rankine cycle is used. The superheated organic vapour from state 6 boiler pressure to state 7 intermediate pressure is expanded in OT_1 which produce work W_{OT1}. Solar energy is used to reheat organic vapour from state 7 to state 8 at constant pressure. Concentrated solar plate collector is used to capture solar energy. Than this reheated organic vapour is expanded in OT_2 from state 8 to state 9 which produce work $W_{OT2.}$ It is assumed that the total pressure drop in organic turbine is equally divided in both organic turbines. In each case the condenser temperature taken to be constant which is 40˚C to ensure proper heat transfer.

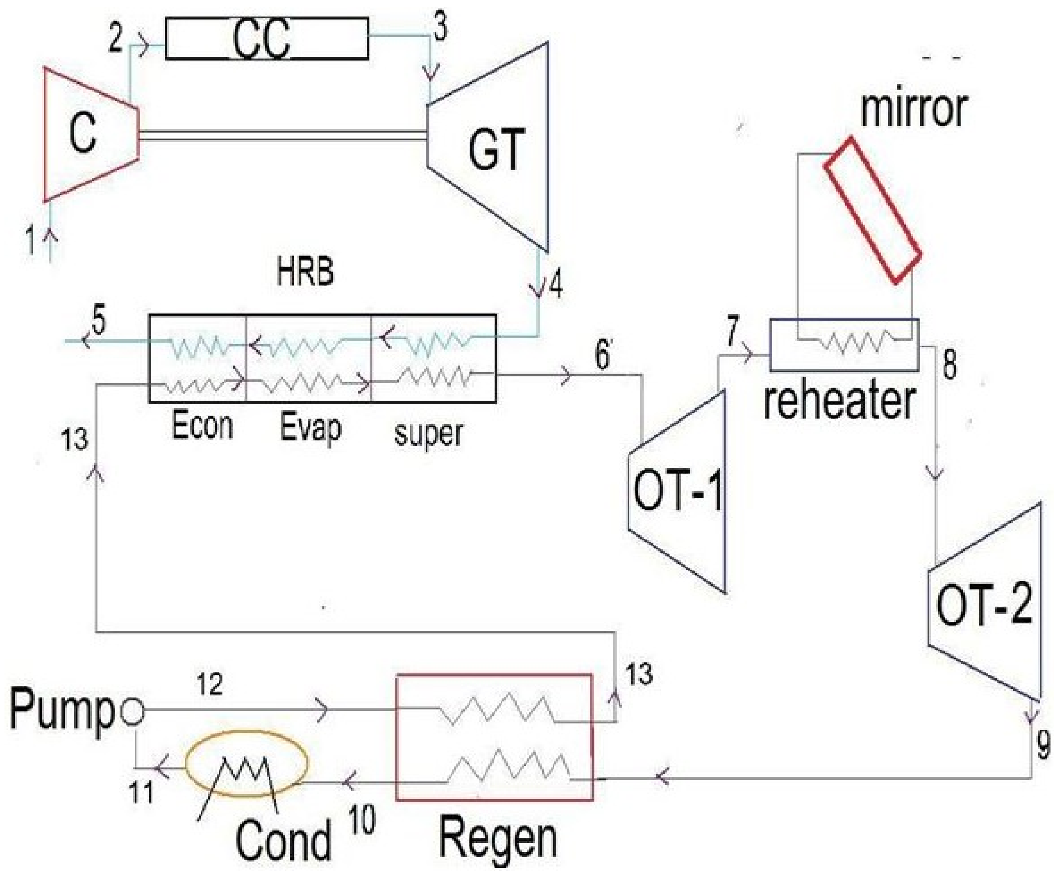

Fig6- combined cycle plant with regeneration and solar reheating

The following organic fluid R134a, R1234yf, R245fa, Acetone is analyzed in this system.

Organic Fluid	**Molecular Mass**	**Condensing Pressure at 40˚C in kpa**	**Normal Boiling Point in ˚C**	**Critical Temperature in ˚C**	**Critical pressure in kpa**
R134a	102	1017	-26.3	100.94	4060
R245fa	134	249.6	15.3	150.86	3651
Acetone	58.08	56.53	56.29	234.85	4700
R1234yf	114	1018	-29.45	94.7	3380

Table1: properties of organic fluids

CHAPTER 4

THERMODYNAMIC ANALYSIS

In the present work, a parametric study with various pressure and temperature at organic turbine inlet has been conducted to determine the efficiency and performance of the organic fluid in the system. The following assumptions are made to simplify the analysis, including energy analysis.

1. All components are assumed to be a steady flow and steady-state process.
2. The changes in the kinetic energy and the potential energy of the components are negligible.
3. The pressure drops and heat loss in the piping connecting the components are negligible.
4. All turbines, compressor, and pump work adiabatically.
5. Pressure drops in HRB, regenerator, and condenser neglected.

4.1 ENERGY ANALYSIS

Based on assumptions, the equations for energy and mass balance are written for each component. Each component in a solar reheated combined cycle plant is shown in Fig.6 considered as control volume.

Mass Balance

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

Energy Balance

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

4.1.1 Energy changes in each component of combined cycle plant:-

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

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

We have compressor efficiency as,

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

Actual compressor work is specified by

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

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 through combustion chamber is given by

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

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

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

We have gas turbine efficiency as,

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

Actual gas turbine work is

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

Heat Recovery Boiler: In HRB energy of hot exhaust gas of gas turbine is utilized for producing superheated vapour of organic fluid. HRB is simply a heat exchanger in which heat transfer is taking place 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,

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

Organic Rankine Turbine: organic turbine is a work producing device in which organic fluid is expanded from boiler pressure to condenser pressure adiabatically. The isentropic work output of ORT,

$$W_{OTi} = \dot{m}_f * (h_6 - h_{7s})$$

We have organic turbine efficiency as,

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

Actual organic turbine work is

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

Condenser: Condenser acts as a heat exchanger in which heat is rejected to environment is given by

$$Qcond = \dot{m}_f * (h10 - h11)$$

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

$$W_{opi} = \dot{m}_f * v_{11} * (P_{12} - P_{11})$$

We have organic pump efficiency as,

$$\eta_{OP} = \frac{W_{OPi}}{W_{OP}}$$

Actual organic pump work is given by

$$W_{op} = \frac{\dot{m}_f * v_{11} * (P_{12} - P_{11})}{\eta_{OP}}$$

Efficiency of Gas Turbine cycle: efficiency of gas turbine plant is given by the ratio of net work output of gas turbine plant and heat supplied in combustion chamber,

$$\eta_{GTP} = \frac{W_T - W_c}{Q_{S1}}$$

Efficiency of Organic Rankine cycle: it is the ratio of net work output of ORC and the total heat supplied in ORC

$$\eta_{ORC} = \frac{W_{OT} - W_{OP}}{Q_{S2}}$$

Efficiency of combined cycle plant: it is defined as the ratio of total work output of the combined cycle plant and total external heat supplied

$$\eta_{CCP} = \frac{(W_T - W_c) + (W_{OT} - W_{OP})}{Q_{S1}}$$

4.1.2 Energy changes in each component of combined cycle plant with regeneration:-

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

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

We have compressor efficiency as

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

Actual compressor work is specified by

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

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 through combustion chamber is given by

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

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

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

We have gas turbine efficiency as,

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

Actual gas turbine work is

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

Heat Recovery Boiler: In HRB energy of hot exhaust gas of gas turbine is utilized for producing superheated vapour of organic fluid. HRB is simply a heat exchanger in which heat transfer is taking place 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,

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

Organic Rankine Turbine: organic turbine is a work producing device in which organic fluid is expanded from boiler pressure to condenser pressure adiabatically. The isentropic work output of ORT,

$W_{OTi} = \dot{m}_f * (h_6 - h_{7s})$

We have organic turbine efficiency as,

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

Actual organic turbine work is

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

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

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

Condenser: Condenser acts as a heat exchanger in which heat is rejected to environment is given by

$Qcond = \dot{m}_f * (h10 - h11)$

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

$W_{opi} = \dot{m}_f * v_{11} * (P_{12} - P_{11})$

We have organic pump efficiency as,

$$\eta_{OP} = \frac{W_{OPi}}{W_{OP}}$$

Actual organic pump work is given by

$$W_{op} = \frac{\dot{m}_f * v_{11} * (P_{12} - P_{11})}{\eta_{OP}}$$

Efficiency of Gas Turbine cycle: efficiency of gas turbine plant is given by the ratio of net work output of gas turbine plant and heat supplied in combustion chamber,

$$\eta_{GTP} = \frac{W_T - W_c}{Q_{S1}}$$

Efficiency of Organic Rankine cycle: it is the ratio of net work output of ORC and the total heat supplied in ORC

$$\eta_{ORC} = \frac{W_{OT} - W_{OP}}{Q_{S2}}$$

Efficiency of combined cycle plant: it is defined as the ratio of total work output of the combined cycle plant and total external heat supplied

$$\eta_{CCP} = \frac{(W_T - W_c) + (W_{OT} - W_{OP)}}{Q_{S1}}$$

4.1.3 Energy changes in each component of combined cycle plant with regeneration and solar reheating:-

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

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

We have compressor efficiency as

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

Actual compressor work is specified by

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

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 through combustion chamber is given by

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

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

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

We have gas turbine efficiency as,

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

Actual gas turbine work is

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

Heat Recovery Boiler: In HRB energy of hot exhaust gas of gas turbine is utilised for producing superheated vapour of organic fluid. HRB is simply a heat exchanger in which heat transfer is taking place 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,

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

Organic Rankine Turbine1: turbine is a work producing device. Organic fluid is expanded from boiler pressure to intermediate pressure in organic turbin-1 adiabatically. The isentropic work output,

$W_{OT1i} = \dot{m}_f * (h_6 - h_{7s})$

We have organic turbine efficiency as,

$$\eta_{OT1} = \frac{h_6 - h_7}{h_6 - h_{7s}}$$

Actual organic turbine work is

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

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

$Q_{S2} = A_s I_s \eta_m$

Heat supplied to organic fluid

$Q_{S2} = \dot{m}_f * (h_8 - h_7)$

Organic Rankine Turbine2: turbine is a work producing device. Organic fluid is expanded from boiler pressure to intermediate pressure in organic turbin-2 adiabatically. The isentropic work output,

$W_{OT2} = \dot{m}_f * (h_8 - h_{9s})$

We have organic turbine efficiency as,

$$\eta_{OT1} = \frac{h_8 - h_9}{h_8 - h_{9s}}$$

Actual organic turbine work is

$W_{OT2} = \dot{m}_f * (h_8 - h_9)$

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

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

Condenser: Condenser acts as a heat exchanger in which heat is rejected to environment is given by

$Q_{cond} = \dot{m}_f * (h_{10} - h_{11})$

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

$W_{OPi} = \dot{m}_f * v_{11} * (P_{12} - P_{11})$

We have organic pump efficiency as,

$$\eta_{OP} = \frac{W_{OPi}}{W_{OP}}$$

Actual organic pump work is given by

$$W_{op} = \frac{\dot{m}_f * v_{11} * (P_{12} - P_{11})}{\eta_{OP}}$$

Efficiency of Gas Turbine cycle: efficiency of gas turbine plant is given by the ratio of net work output of gas turbine plant and heat supplied in combustion chamber,

$$\eta_{GTP} = \frac{W_{GT} - W_C}{Q_{S1}}$$

Efficiency of Organic Rankine cycle: it is the ratio of net work output of ORC and the total heat supplied in ORC

$$\eta_{ORC} = \frac{W_{OT} + W_{OT2} - W_{OP}}{Q_{S2} + Q_{HRB}}$$

Efficiency of combined cycle plant: it is defined as the ratio of total work output of the combined cycle plant and total external heat supplied

$$\eta_{CCP} = \frac{(W_{GT} - W_C) + (W_{OT1} + W_{OT2} - W_{OP})}{Q_{S1} + Q_{S2}}$$

4.2 EXERGY ANALYSIS

Exergy destruction is given by

$$\mathbf{ED = T_0 * \dot{m}_a * (\Delta s_{sys} + \Delta s_{surr})}$$

4.2.1 Exergy analysis of each component of combined cycle plant without regeneration-

"Exergy destruction of compressor"

$$\mathbf{ED_C = \dot{m}_a * T_0 * (s_2 - s_1)}$$

"Exergy destruction of gas turbine"

$$\mathbf{ED_{GT} = \dot{m}_a * T_0 * (s_4 - s_3)}$$

"Exergy destruction in Heat Recovery Boiler"

$$\mathbf{ED_{HRB} = \dot{m}_a * T_0 * (s_5 - s_4) + \dot{m}_f * T_0 * (s_6 - s_{13})}$$

"Exergy destruction in organic turbine"

$$\mathbf{ED_{OT} = \dot{m}_f * T_0 * (s_7 - s_6)}$$

"Exergy destruction in condenser"

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

"Exergy destruction in organic Rankine cycle pump"

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

"Exergy transfer in gas turbine cycle by combustion chamber"

ET_{CC} = Q_{S1} - $\dot{m}_a$*T_0*(s_3-s_2)

"Exergy transfer by HRB to organic fluid"

ET_{HRB} = Q_{HRB} - $\dot{m}_f$*T_0*(s_6-s_{13})

“Second law efficiency of gas turbine power plant"

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

"Second law efficiency of organic Rankine cycle"

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

"Second law efficiency of combined cycle plant"

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

4.2.2 Exergy analysis of each component of combined cycle plant with regeneration-

"Exergy destruction of compressor"

ED_C = $\dot{m}_a$*T_0*(s_2-s_1)

"Exergy destruction of gas turbine"

ED_{GT} = $\dot{m}_a$*T_0*(s_4-s_3)

"Exergy destruction in Heat Recovery Boiler"

ED_{HRB} = $\dot{m}_a$*T_0*(s_5-s_4) + $\dot{m}_f$*T_0*(s_6-s_{13})

"Exergy destruction in organic turbine"

ED_{OT} = $\dot{m}_f$*T_0*(s_7-s_6)

“Exergy Destruction in regenerator”

$ED_{REG} = \dot{m}_f*T_0*(s_{10}-s_9) + \dot{m}*T_0*(s_{13}-s_{12})$

"Exergy destruction in condenser"

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

"Exergy destruction in organic Rankine cycle pump"

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

"Exergy transfer in gas turbine cycle by combustion chamber"

$ET_{CC} = Q_{S1} - \dot{m}_a*T_0*(s_3-s_2)$

"Exergy transfer by HRB to organic fluid"

$ET_{HRB} = Q_{HRB} - \dot{m}_f*T_0*(s_6-s_{13})$

“Second law efficiency of gas turbine power plant"

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

"Second law efficiency of organic Rankine cycle"

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

"Second law efficiency of combined cycle plant"

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

4.2.3 Exergy analysis of each component of combined cycle plant with regeneration and solar reheating-

"Exergy destruction of compressor"

$ED_C = \dot{m}_a*T_0*(s_2-s_1)$

"Exergy destruction of gas turbine"

$ED_{GT} = \dot{m}_a*T_0*(s_4-s_3)$

"Exergy destruction in Heat Recovery Boiler"

ED_{HRB} = $\dot{m}_a$*T_0*(s_5-s_4) + $\dot{m}_f$*T_0*(s_6-s_{13})

"Exergy destruction in organic turbine-1"

ED_{OT1} = $\dot{m}_f$*T_0*(s_7-s_6)

"Exergy destruction in solar reheater"

$ED_{reheater}$ = $\dot{m}_f$*T_0*(s_8-s_7) – Q_{S2}*(T_0/T_s)

"Exergy destruction in organic turbine-2"

ED_{OT2} = $\dot{m}_f$*T_0*(s_9-s_8)

"Exergy Destruction in regenerator"

ED_{REG} = $\dot{m}_f$*T_0*(s_{10}-s_9) + $\dot{m}_f$*T_0*(s_{13}-s_{12})

"Exergy destruction in condenser"

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

"Exergy destruction in organic Rankine cycle pump"

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

"Exergy transfer in gas turbine cycle by combustion chamber"

ET_{CC} = Q_{S1} - $\dot{m}_a$*T_0*(s_3-s_2)

"Exergy transfer by HRB to organic fluid"

ET_{HRB} = Q_{HRB} - $\dot{m}_f$*T_0*(s_6-s_{13})

"Exergy transfer by solar heater to organic fluid"

$ET_{reheater}$ = Q_{S2} - $\dot{m}_f$*T_0*(s_8-s_7)

"Second law efficiency of gas turbine power plant"

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

"Second law efficiency of organic Rankine cycle"

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

"Second law efficiency of combined cycle plant"

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

4.3 Input Parameters:

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

Air compressor inlet temperature	T_1= 27˚C
Air compressor inlet pressure	P_1= 100 kPa
Pressure ratio of compressor	r_P= 18
Isentropic efficiency of air compressor	η_C= 0.85
Isentropic efficiency of Gas turbine	η_{GT}= 0.85
Power output of Gas Turbine Plant	W_{GTP}= 25MW
Pinch point temperature difference	ΔT_{pinch}= 20˚C
Organic Rankine turbine inlet pressure	P_1= 2500 to 4000 kPa
Isentropic efficiency of organic turbine	η_{OT}= 0.85
Solar irradiation on CSP collector	I_S= 900 W/m^2
Mirror efficiency of CSP collector	η_m= 0.70
Effectiveness of regenerator LVHE	e = 0.90
Condenser Temperature	Tc = 40˚C
Efficiency of organic Rankine pump	η_{OP}= 0.90
Dead State Temperature	To= 250C
Organic fluid used	R134a, R245fa, R1234yf, and Acetone

CHAPTER 5

RESULTS AND DISCUSSION

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

5.1 Comparisons of without Regeneration, with Regeneration, and Regeneration with Solar reheat cycle:

Figure 7 to 23 shows comparison of efficiency against the organic turbine inlet temperature and pressure. From the figures it is clear that the efficiency of organic Rankine cycle increases with regeneration while the efficiency of combined cycle plant remains almost constant. By using regeneration heat supplied to organic cycle plant decreases hence rate of evaporation increase which results output increases and hence organic Rankine cycle efficiency increases.

With solar reheating organic Rankine cycle plant shows increase in efficiency while combined cycle plant shows decrease in efficiency because of decrease in mean temperature of heat addition in the system with superheating..

5.2 Comparison of various organic fluids:

Figure 24 to 29 shows variation of efficiencies of the system at organic turbine inlet temperature, pressure, effectiveness of regenerator and solar collector mirror area. Among all organic fluid Acetone shows maximum efficiency of organic Rankine cycle which is about 25.96% and combined cycle has efficiency about 54.6%. R134a, R245fa, R1234yf and Acetone have maximum organic Rankine cycle efficiency of 13.31%, 21.44%, 13.65% and 25.96% respectively at 2500kPa maximum pressure and 200˚C maximum temperature of ORC fluid.

5.3 Exergy efficiency:

Variation of Exergy efficiency of various organic fluids with maximum temperature, maximum pressure of organic cycle is shown in figure 30 to 35. It is seen that Exergy efficiency of organic Rankine cycle increase with maximum temperature and maximum pressure. It is seen that efficiency of combined cycle plants decreases with increase in organic Rankine cycle maximum temperature but increases with increase in maximum pressure.

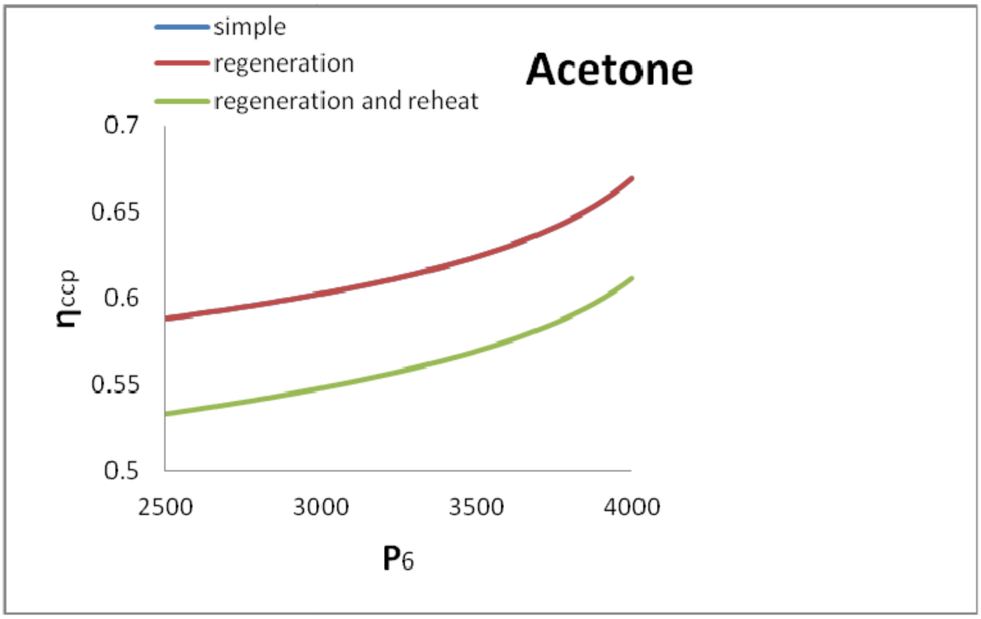

Fig7: variation of efficiency of combined cycle plant with organic Rankine cycle maximum pressure of Acetone

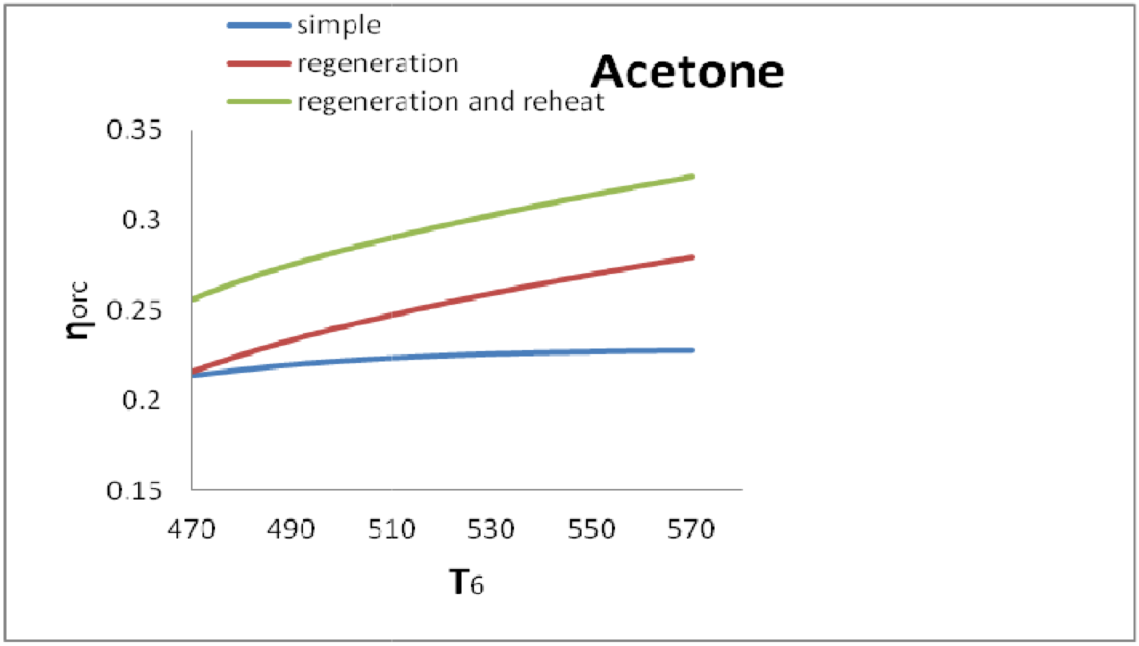

Fig8 variation of efficiency organic Rankine cycle with organic Rankine cycle maximum temperature of Acetone

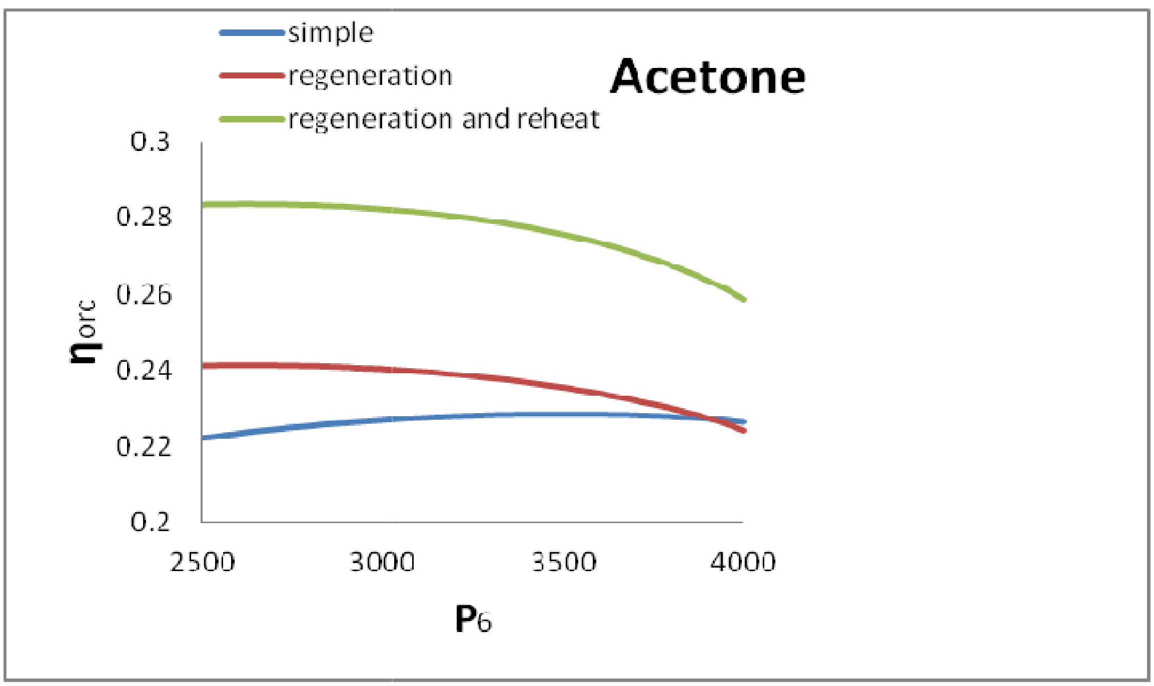

Fig9: variation of efficiency organic Rankine cycle with organic Rankine cycle maximum pressure of Acetone

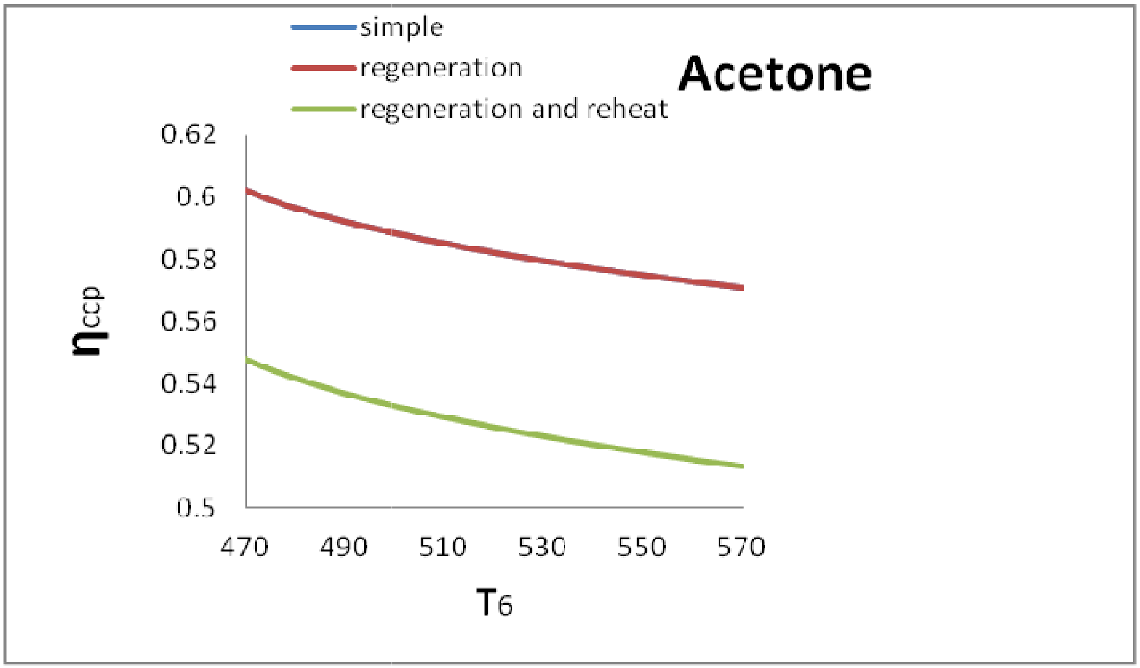

Fig10: variation of efficiency of combined cycle plant with organic Rankine cycle maximum temperature of Acetone

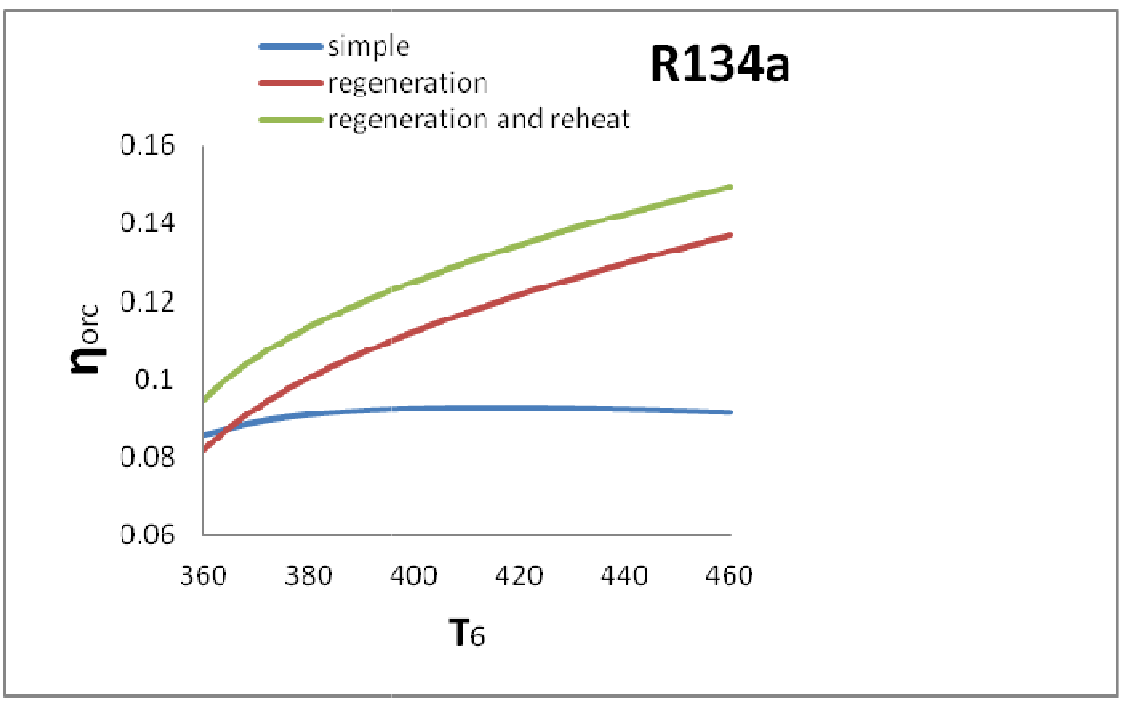

Fig11: variation of efficiency of organic Rankine cycle plant with organic Rankine cycle maximum Temperatur of R134a

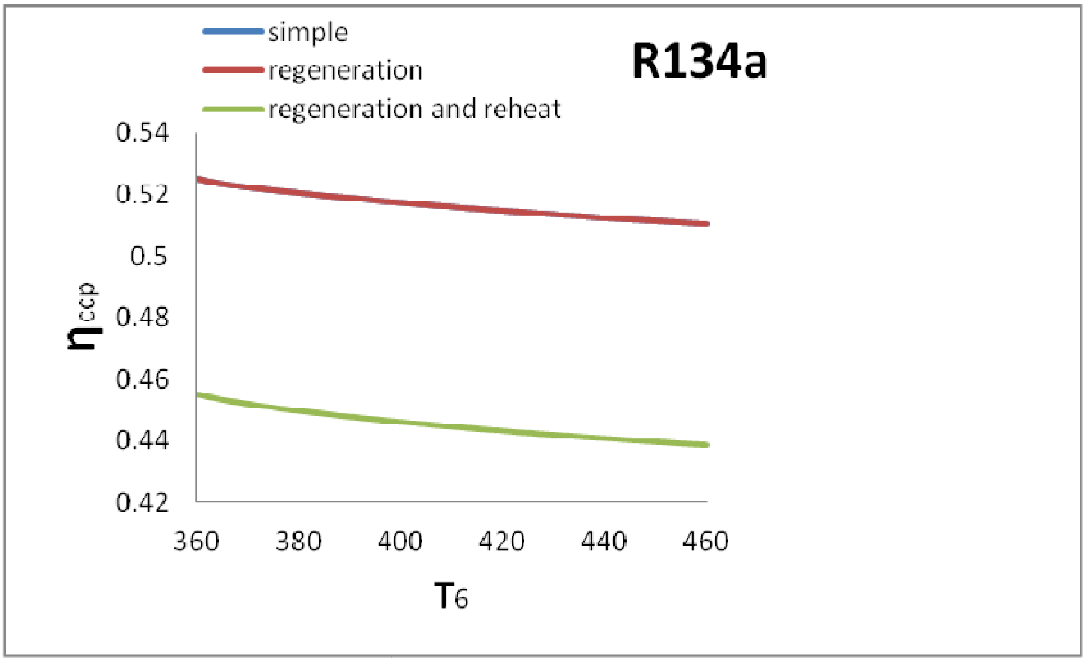

Fig12: variation of efficiency of combined cycle plant with organic Rankine cycle maximum Temperature of R134a

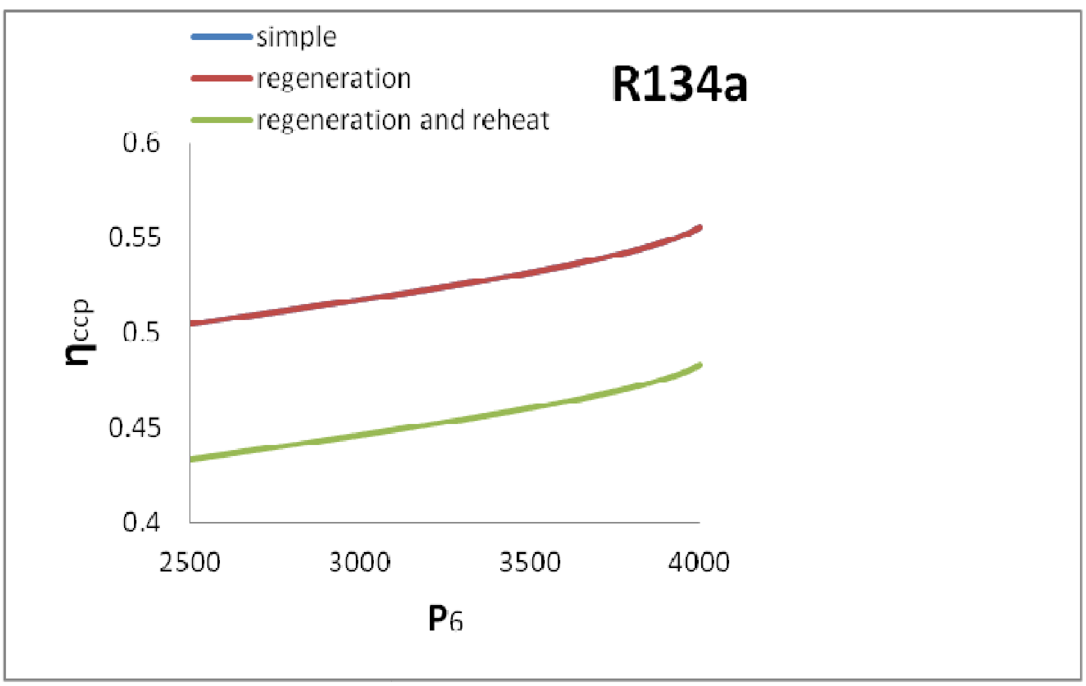

Fig13: variation of efficiency of combined cycle plant with organic Rankine cycle maximum Pressure of R134a

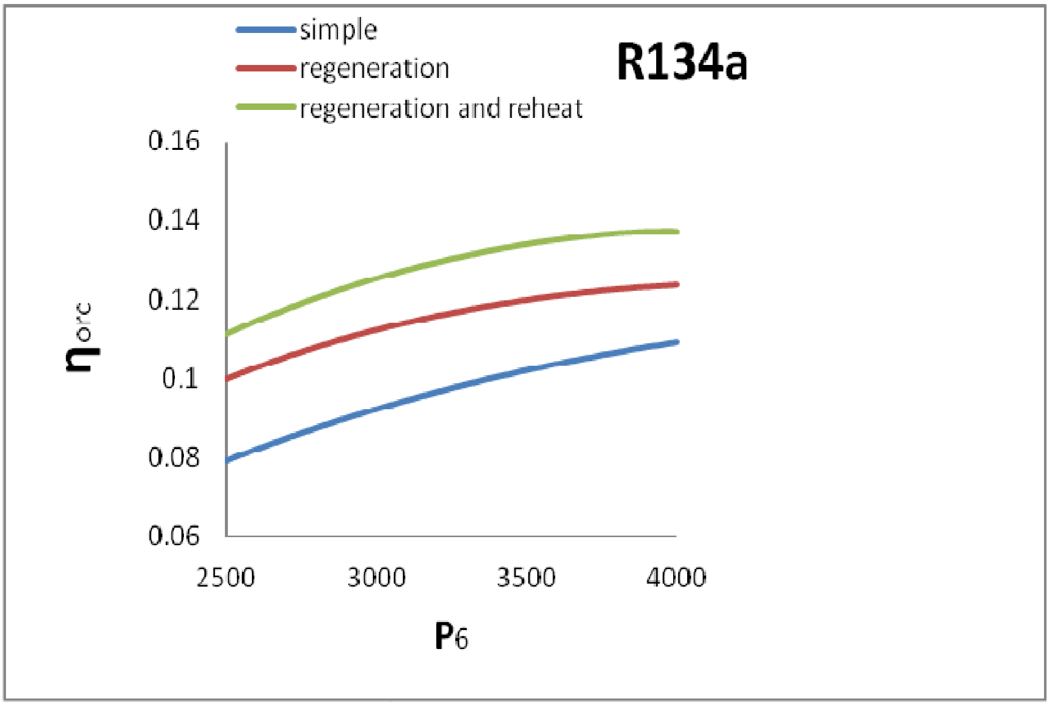

Fig14: variation of efficiency of organik Rankine cycle with organic Rankine cycle maximum Pressure of R134a

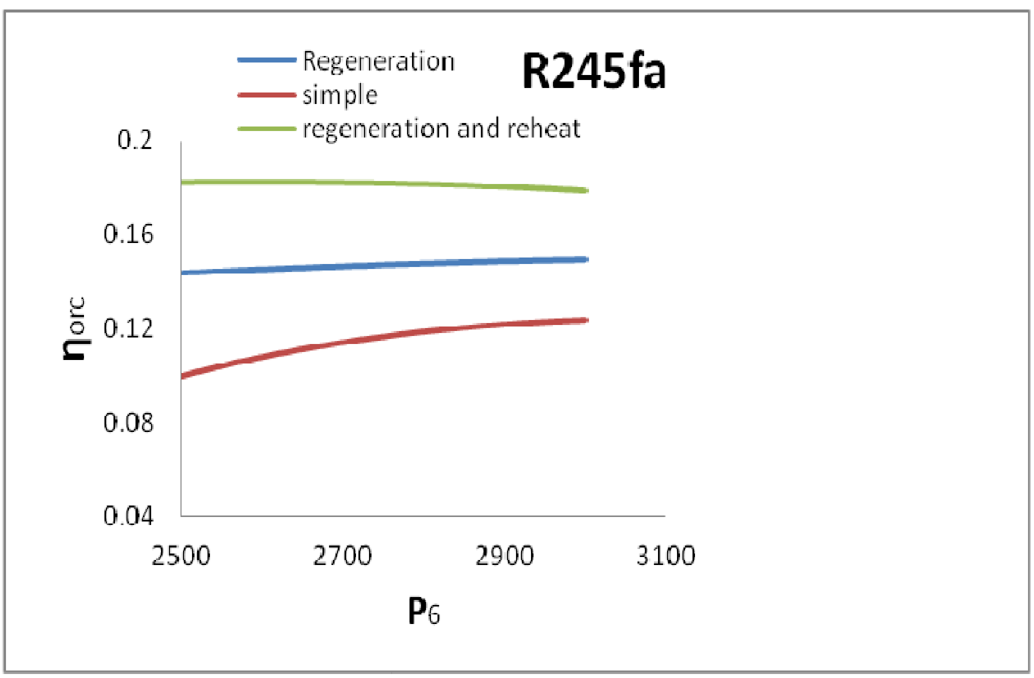

Fig15: variation of efficiency of organik Rankine cycle plant with organic Rankine cycle maximum Pressure of R245fa

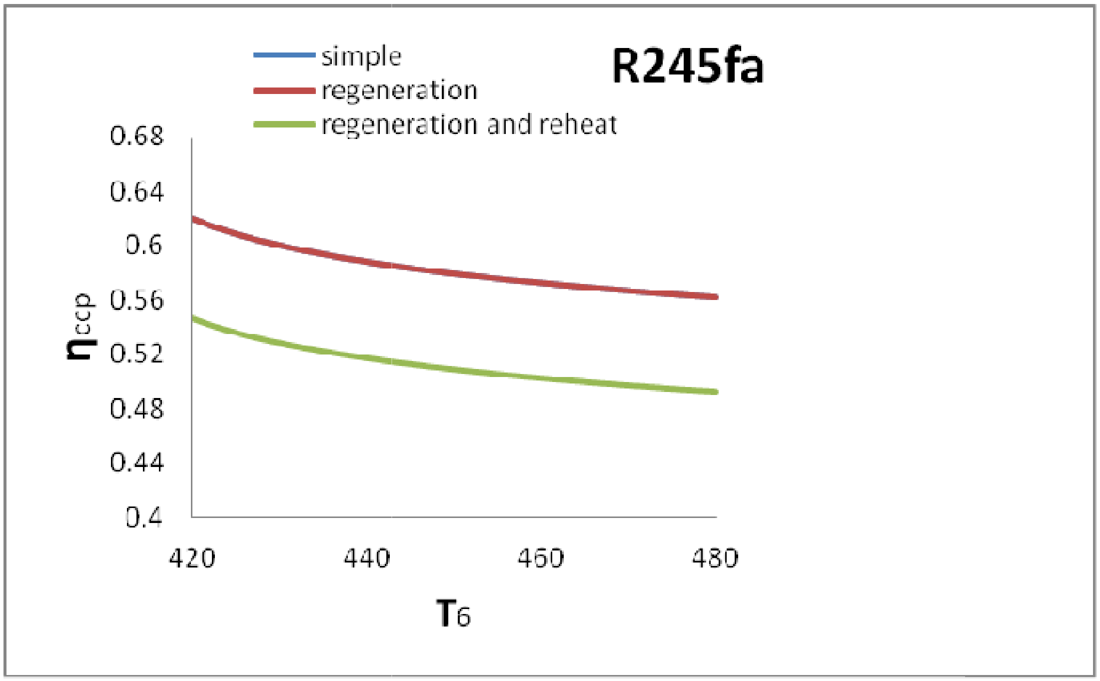

Fig16: variation of efficiency of combined cycle plant with organic Rankine cycle maximum Temperature of R245fa

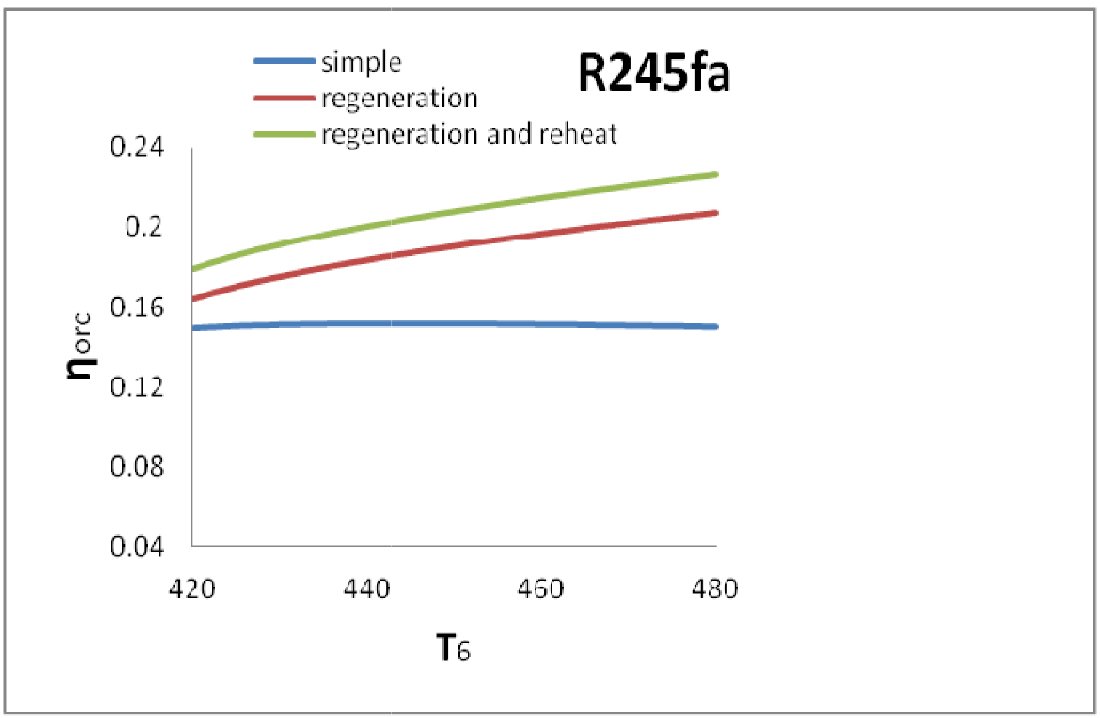

Fig17: variation of efficiency of organik Rankine cycle plant with organic Rankine cycle maximum Temperature of R245fa

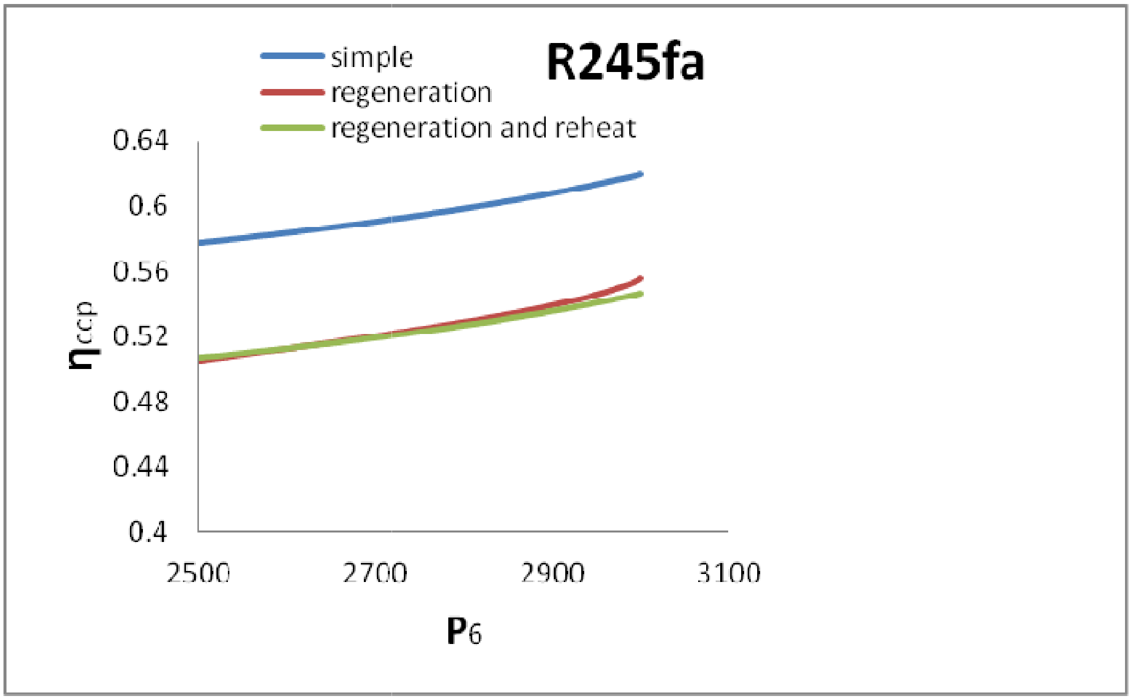

Fig18: variation of efficiency of combined cycle plant with organic Rankine cycle maximum pressure of R245fa

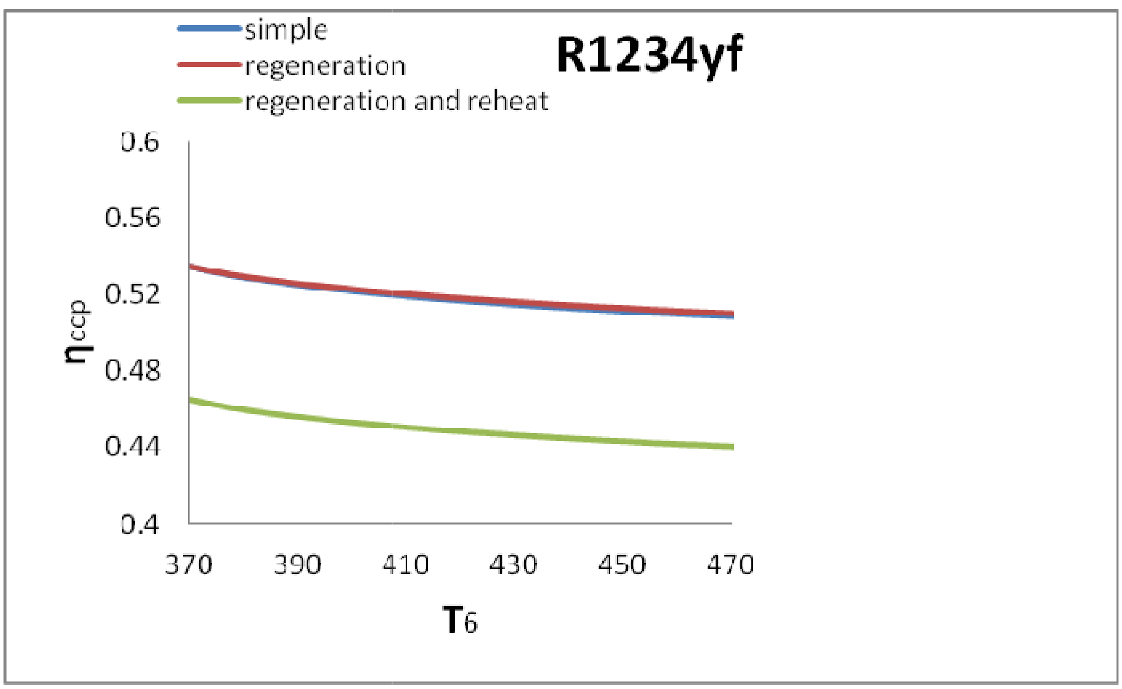

Fig19: variation of efficiency of combined cycle plant with organic Rankine cyclemaximum Temperature of R1234yf

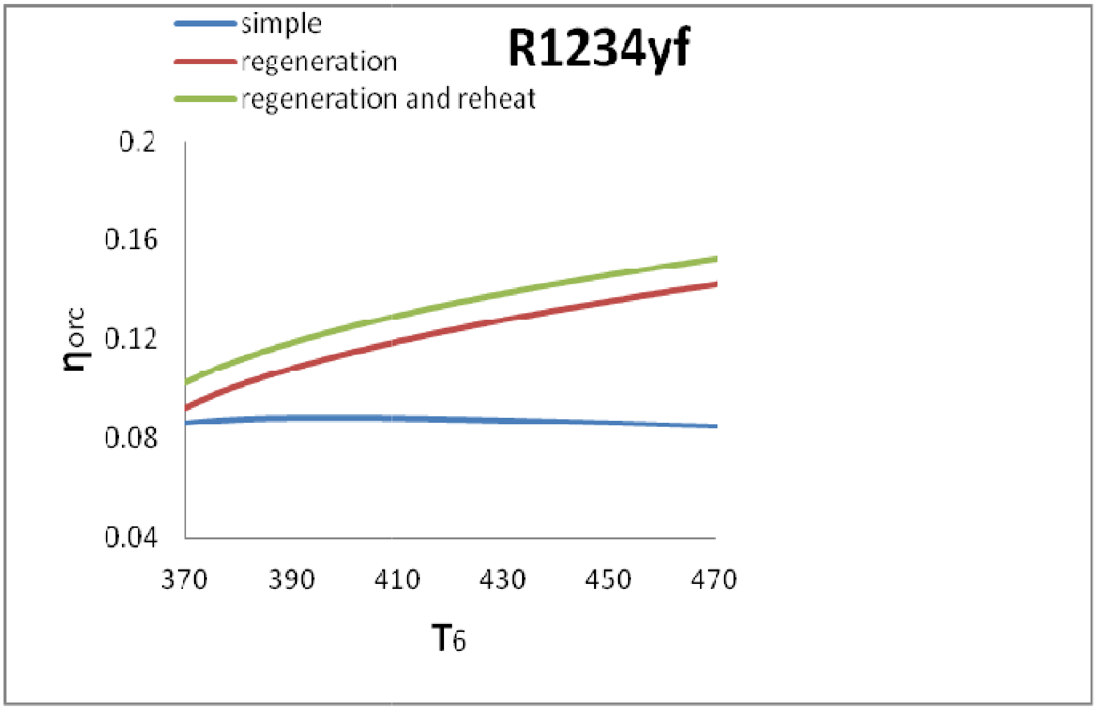

Fig 20: variation of efficiency of organic Rankine cycle with organic Rankine cycle maximum temperature of R1234yf

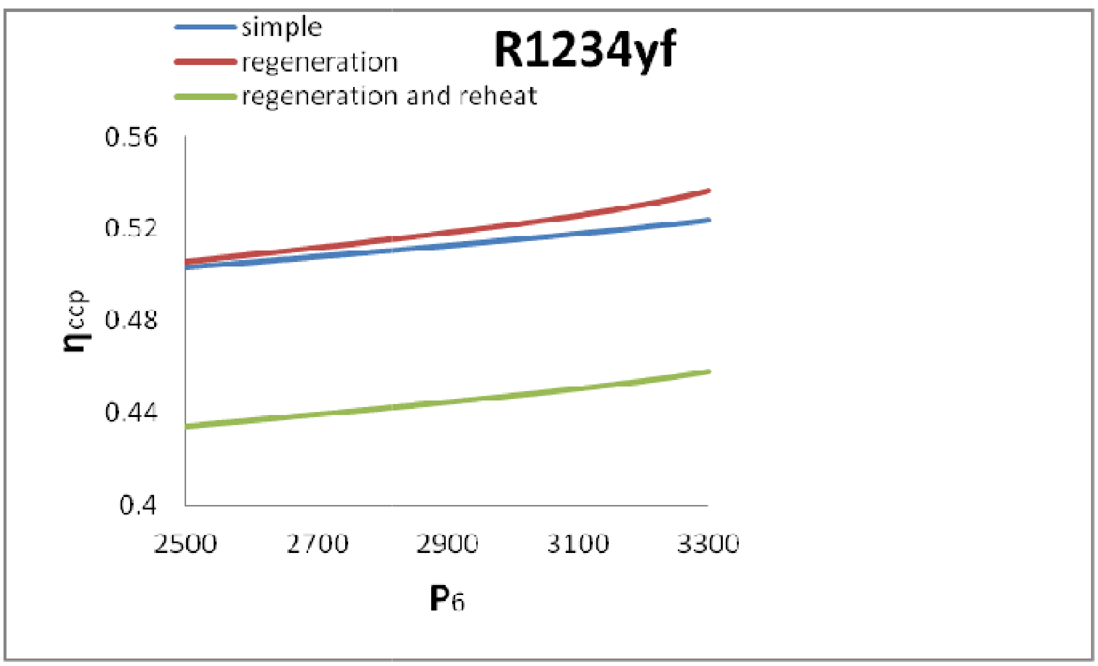

Fig 21: variation of efficiency of combined cycle plant with organic Rankine cycle maximum Pressure of R1234yf

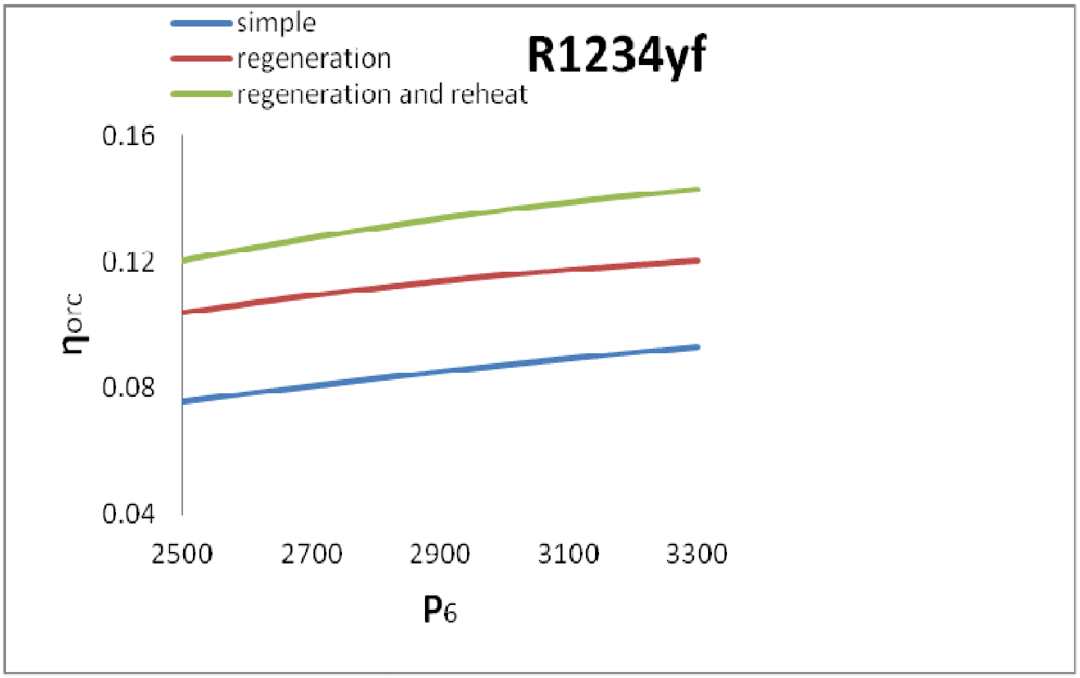

Fig 22: variation of efficiency of organik Rankine cycle plant with organic Rankine cycle maximum Pressure of R1234yf

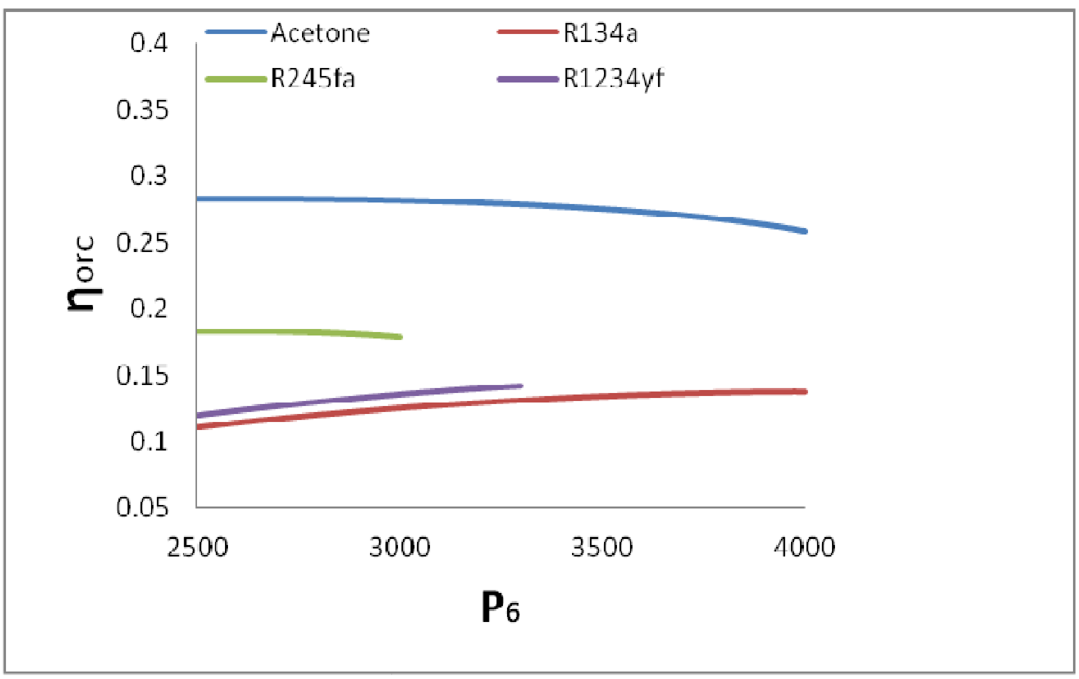

Fig 23: variation of efficiency of organik Rankine cycle plant with organic Rankine cycle maximum Pressure

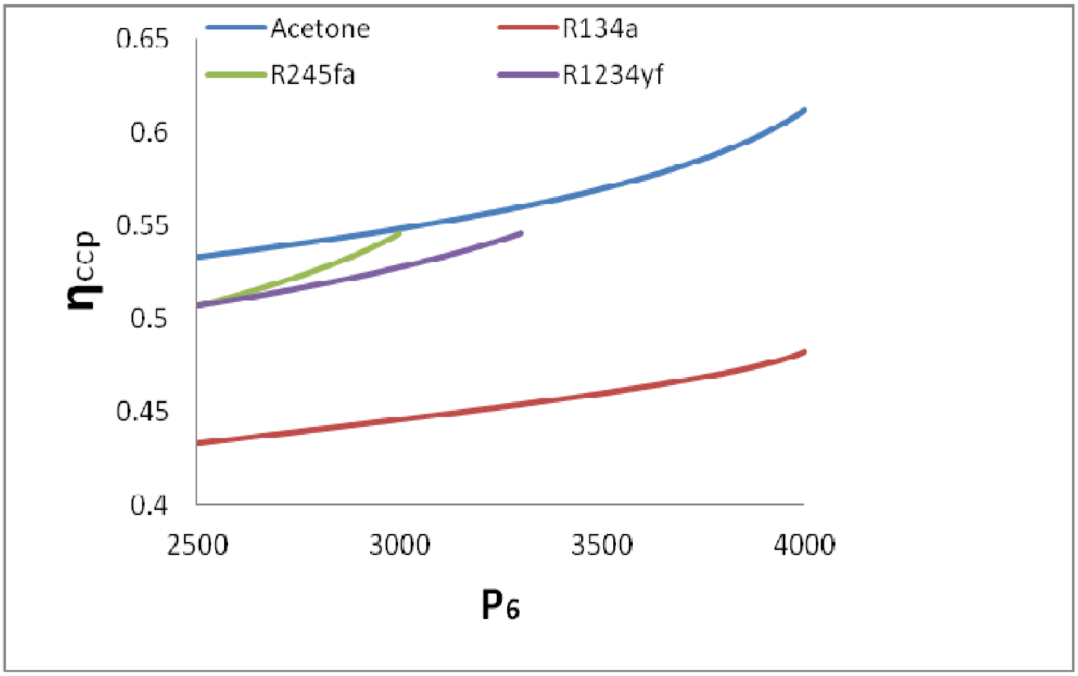

Fig 24: variation of efficiency of combined cycle plant with organic Rankine cycle maximum Pressure

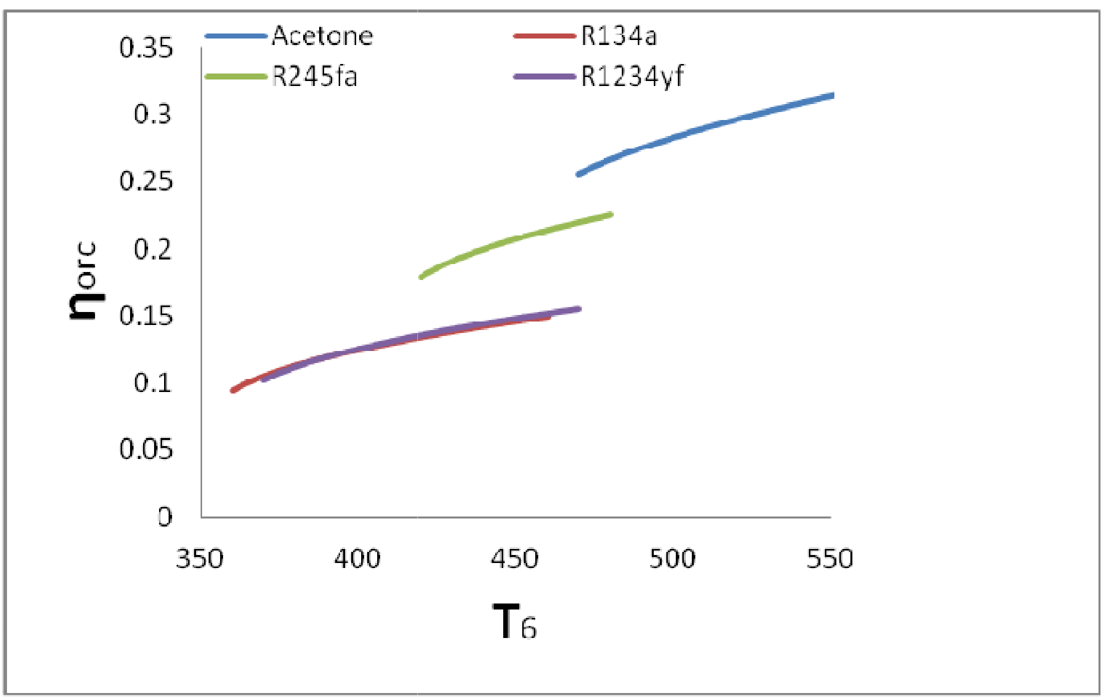

Fig 25: variation of efficiency of organik Rankine cycle plant with organic Rankine cycle maximum Temperature

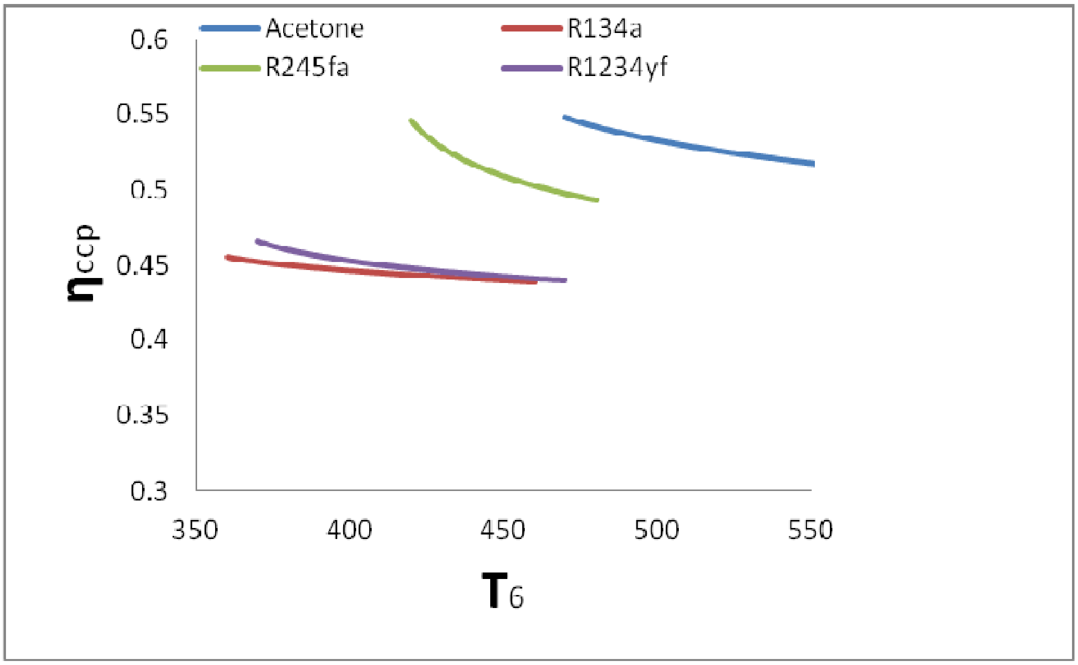

Fig 26: variation of efficiency of combined cycle plant with organic Rankine cyclemaximum Temperature

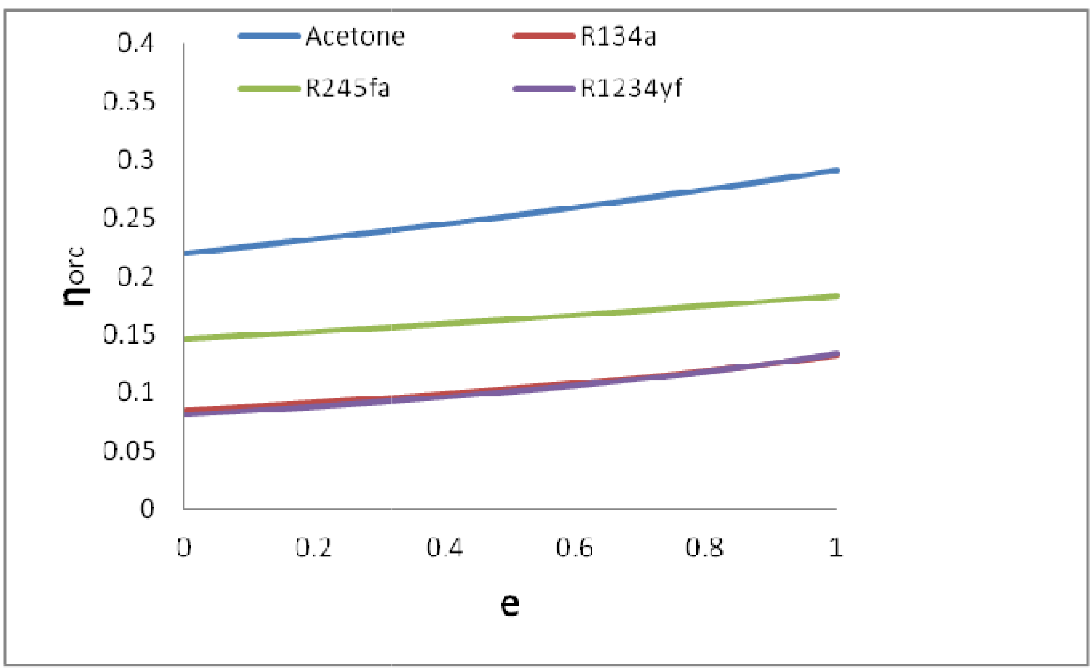

Fig 27: variation of efficiency of organik Rankine cycle plant with effectiveness of regenerator

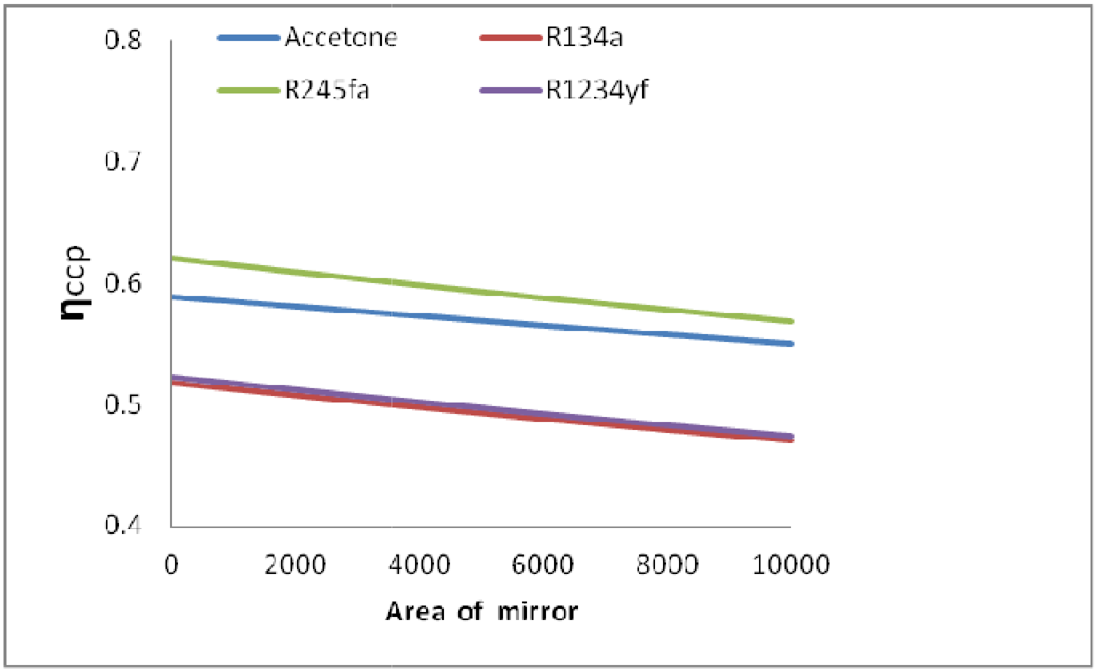

Fig 28: variation of efficiency of combined cycle plant with solar reheater mirror area

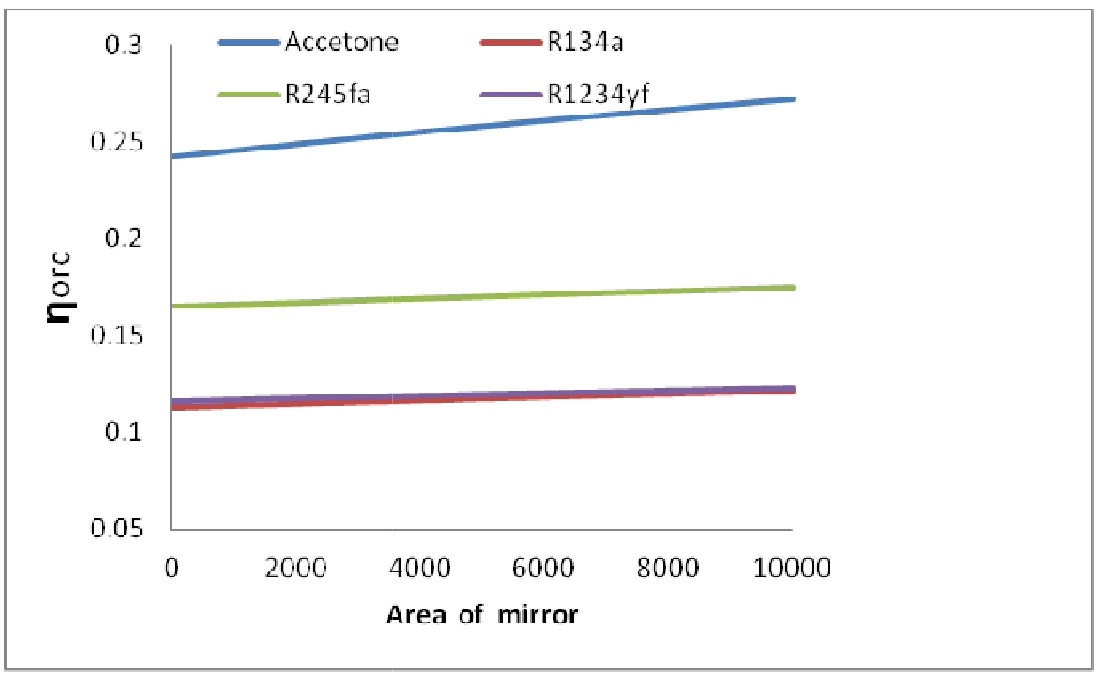

Fig 29: variation of efficiency of organic Rankine cycle plant with solar reheater mirror area

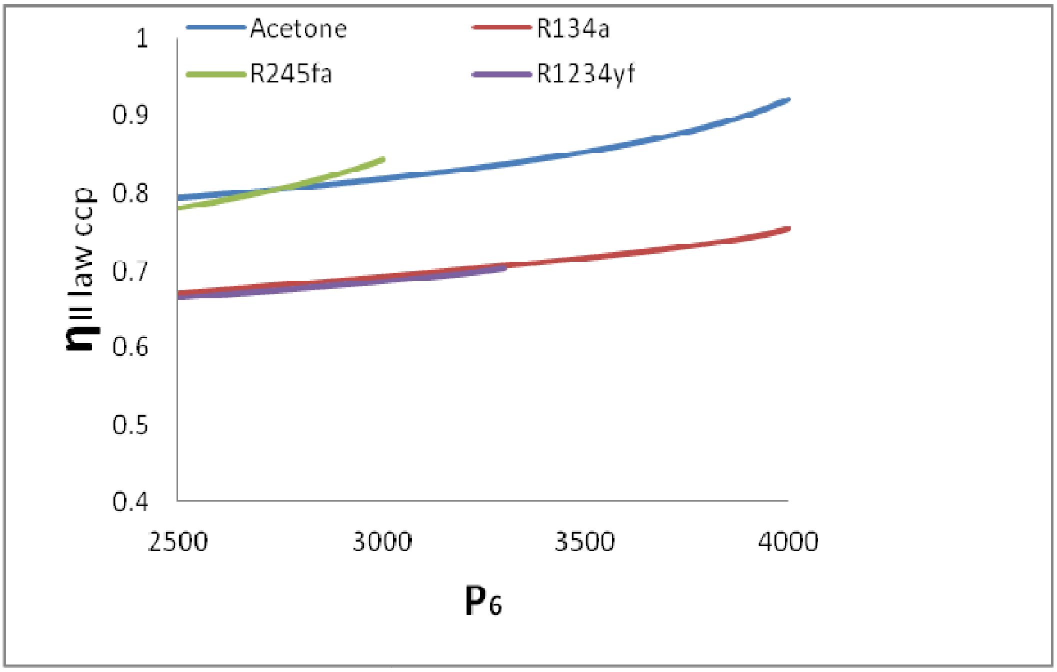

Fig 30: variation of second law efficiency of combined cycle plant with organic Rankine maximum pressure

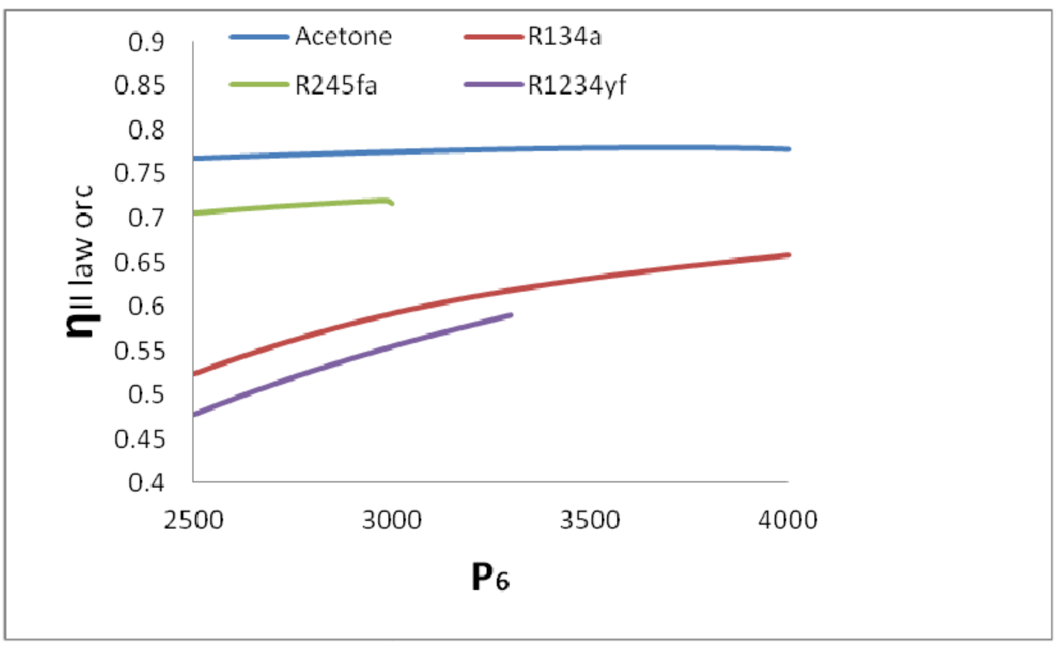

Fig 31: variation of second law efficiency of organic Rankine plant with organic Rankine maximum pressure

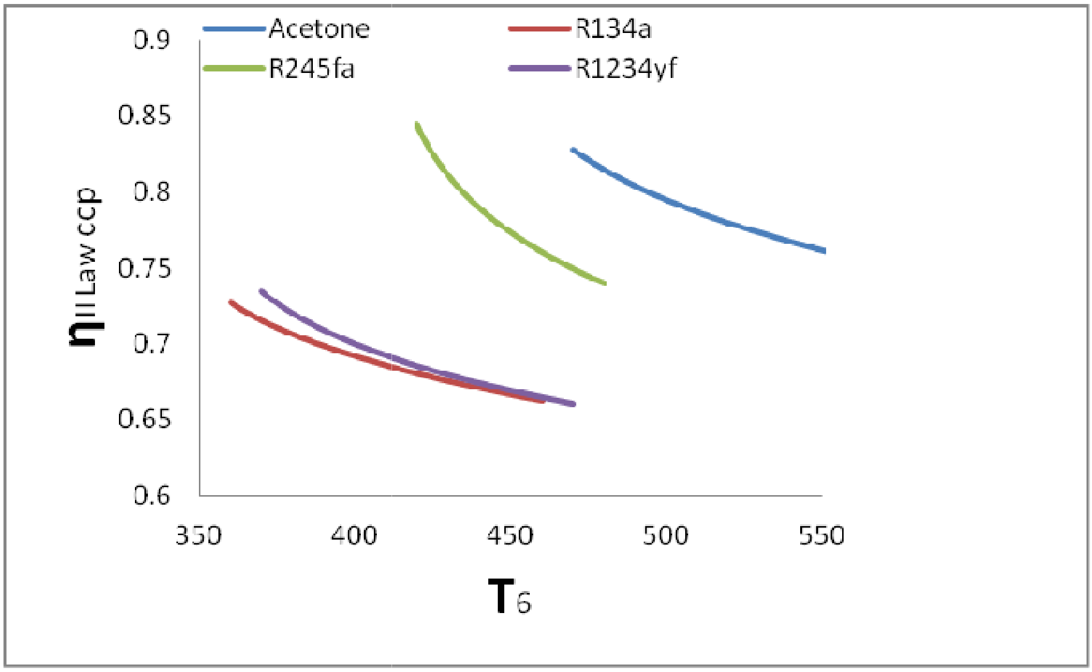

Fig 32: variation of second law efficiency of combined cycle plant with organic Rankine maximum Temperature

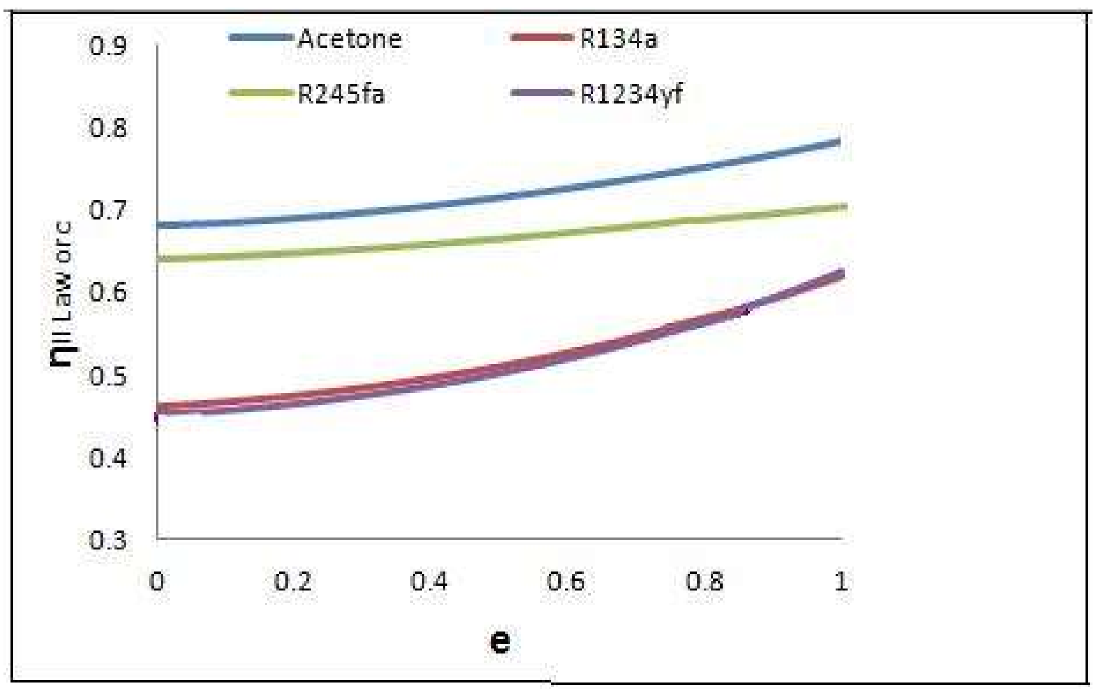

Fig 33 variation of second law efficiency of organic Rankine cycle with effectiveness of regenerator

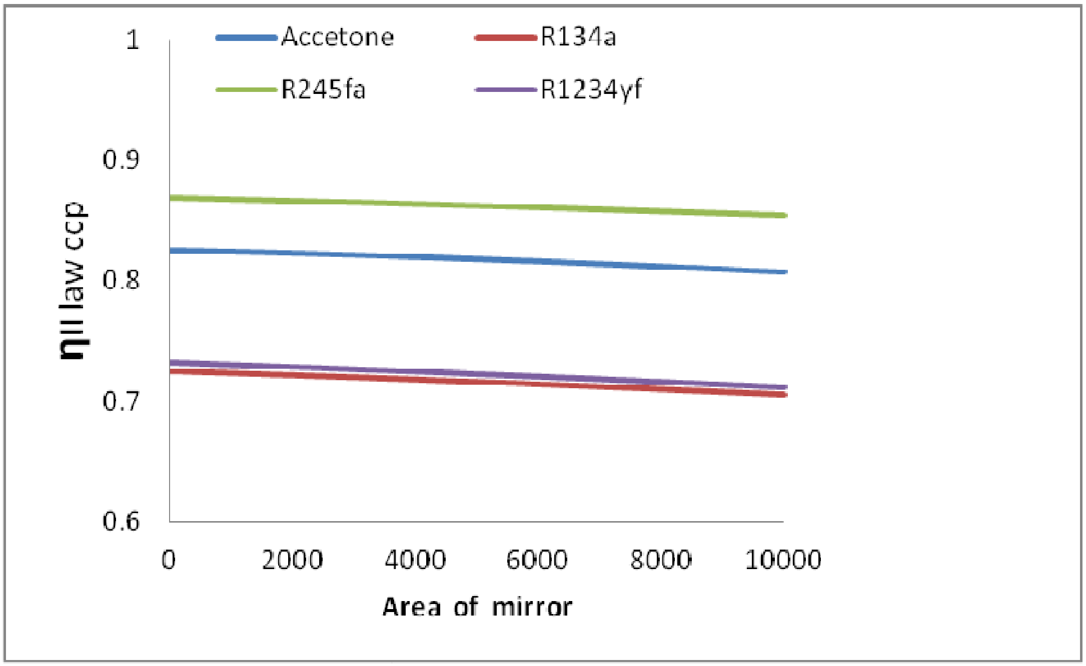

Fig34: variation of second law efficiency of combined cycle plant with solar reheater mirror area

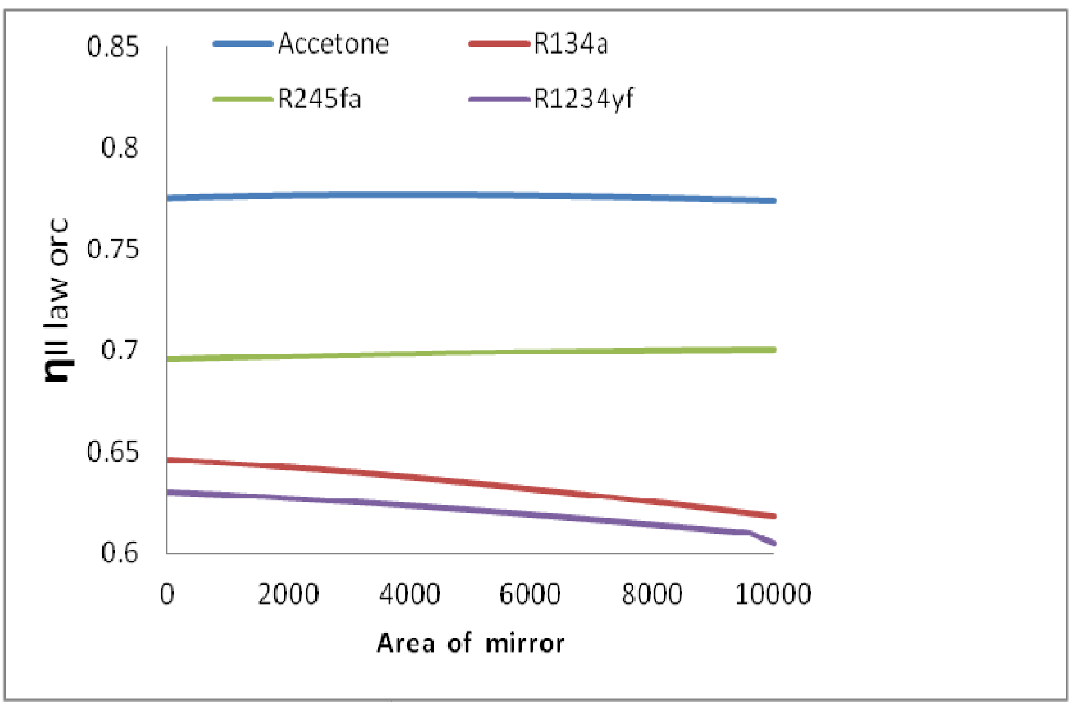

Fig35: variation of second law efficiency of organic Rankine cycle plant with solar reheater mirror area

Efficiency of cycle at organic Rankine cycle maximum pressure 2500kPa and maximum temperature 200˚C.

- **Efficiency of Combined cycle plant:**

	R134a	**$R245fa$**	**R1234yf**	**Acetone**
Without Regeneration	49.97	55.02	49.18	60.03
With Regeneration	49.97	55.02	49.18	60.03
Regeneration and Solar Reheating	43.14	48.16	42.94	54.6

Table2: Efficiency of Combined cycle plant for different organic fluid

- **Efficiency of Organic Rankine cycle:**

	R134a	**R245fa**	**R1234yf**	**Acetone**
Without Regeneration	7.68	14.20	7.21	21.53
With Regeneration	12.30	19.63	12.79	21.90
Regeneration and Solar reheating	13.33	21.44	13.65	25.96

Table3: Efficiency of organic Rankine cycle for different organic fluid

- **Second law efficiency of combined cycle plant with Regeneration and solar Reheat**:

	R134a	**R245fa**	**R1234yf**	**Acetone**
$\eta_{II\ law\ ccp}$	64.57	72.34	64.33	82.34

Table 4: Second law efficiency of Regeneration and solar Reheat system

- **Second law efficiency of organic Rankine cycle plant with Regeneration and solar Reheat**:

	R134a	**R245fa**	**R1234yf**	**Acetone**
$\eta_{II\ law\ orc}$	38.65	66.05	38.32	77.27

Table 5: Second law efficiency of ORC with Regeneration and solar Reheat system

CHAPTER 6

CONCLUSION

In this thesis, an extensive first law (energy) and second law (Exergy) analysis of R134a, R245fa, R1234yf, and Acetone Organic fluids in combined cycle with Regeneration and Reheating is presented. 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. Acetone have higher first law efficiency and Exergetic efficiency (second law efficiency) in Organic Rankine cycle but R1234yf have higher first law efficiency and 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. Acetone is recommended for practical applications due to its highest exergetic efficiency among selected organic fluid but some important problem related to flammability and explosion risk have to be considered while managing it.

7. R245fa has highest exergetic efficiency after Acetone, compared to remaining selected organic fluids and it is recommended for practical applications after Acetone.

CHAPTER 7

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