THERMODYNAMIC (ENERGY AND EXERGY) ANALYSIS OF SOLAR ASSISTED POWER COOLING COMBINED GENERATION SYSTEMS

THESIS

SUBMITTED TO DELHI TECHNOLOGICAL UNIVERSITY

FOR THE AWARD OF THE DEGREE OF

DOCTOR OF PHILOSOPHY

IN

MECHANICAL ENGINEERING

By

KAUSHALENDDRA KUMAR DUBEY

DEPARTMENT OF MECHANICAL ENGINEERING

DELHI TECHNOLOGICAL UNIVERSITY

DELHI-110042 (INDIA)

AUGUST, 2020

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KAUSHALENDDRA KUMAR DUBEY

[Roll No. 2K13/Ph.D. ME/02]

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"It is through the blessings of God that I have been able to complete my thesis and He alone, is worthy of all praise"

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Kaushalendra Kumar Dubey

DELHI TECHNOLOGICAL UNIVERSITY, DELHI

DECLARATION

I hereby declare that the research work presented in this thesis entitled **"THERMODYNAMIC (ENERGY AND EXERGY) ANALYSIS OF SOLAR ASSISTED POWER COOLING COMBINED GENERATION SYSTEMS"** is an original work carried out by me under the supervision of Prof. R. S. Mishra, Professor, Department of Mechanical Engineering, Delhi Technological University, Delhi. This thesis has been prepared in conformity with the rules and regulations of the Delhi Technological University, Delhi. The research work reported and results presented in the thesis have not been submitted either in part or full to any other university or institute for the award of any other degree or diploma.

Kaushalendra Kumar Dubey

DELHI TECHNOLOGICAL UNIVERSITY, DELHI

CERTIFICATE

This is to certify that the work embodied in the thesis entitled **THERMODYNAMIC (ENERGY AND EXERGY) ANALYSIS OF SOLAR ASSISTED POWER COOLING COMBINED GENERATION SYSTEMS"** by **Kaushalendra Kumar Dubey, (Roll No: 2K13/Ph.D. ME/02)** in partial fulfillment of requirements for the award of Degree of **DOCTOR OF PHILOSOPHY in Mechanical Engineering**, is an authentic record of student's own work carried by him under my supervision.

This is also certified that this work has not been submitted to any other Institute or University for the award of any other diploma or degree.

Prof. (Dr.) R. S. Mishra
Professor
Department of Mechanical Engineering
Delhi Technological University
Delhi

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ABSTRACT

Abstract

The worldwide power industry structure is changing to a market economy to ensure commercial availability. This captive industry is essential for national development power & electricity infrastructure, but more than 40% of the input energy is lost during the plant operation with different thermal utilities. These precious dumped or waste heats have tremendous potential for the generation of multiple effects of energy like heating-power & cooling and also help to enhance the efficiency of thermodynamic power cycles. The novel and advanced thermodynamic systems are important because solar based on refrigeration systems have been discussed in the proposed title of research for different categories of waste heat source recovery. Therefore, this study investigates both theoretical and software-based simulation into the distinguishing feature of the advanced concept of the Rankine model called Organic Rankine Cycle (ORCs), which has three useful output heating-power and cooling production. The present research work focuses on two significant thermodynamic analyses: the integration of advanced thermodynamic cycles and the heat recovery system with employment of solar thermal systems, and the second is complete thermodynamic analysis, consisting of Energy and Exergy analysis. The traditional approach of thermodynamics analysis is based on the 1st law and 2nd law of thermodynamics. The 1st law of thermodynamics (FLT) gives the work, heat transfer, energy performance, thermal efficiency. In contrast, the 2nd law of thermodynamics (SLT) provides the system's actual performance by entropy generation (exergy) principle. The quantity and quality of energy (useful energy) are estimated through the FLT and SLT. The main objective of 2E analysis (energy-exergy) is to examine the theoretical and actual performance of proposed thermal systems to identify energy loss in integrated parts of the thermal system for efficient performance.

The conventional mathematical modeling is suitable for physical system simulation, complex mathematical model development, and quantitative analysis. The statistical modeling helps to error estimation, system optimization, and comparative study of fundamental & predicted complex analysis results. Several statistical analysis methods are available with new artificial intelligence applications like linear or multi-linear regression method, artificial neural network method, least square method, and Taguchi-Annova

method. All optimization techniques are suitable for the least parameter identification, error count with complex problem-solving. This

Research work is referred to as the multi-linear regression method for parametric identification and actual- predicted result comparisons.

This research work consisted of four thermodynamics models for combined cooling, heating, and power generation effect using waste heat of different power plants.

1. Stack Flow heat recovery of Combined GT-ST plant using the LiBr-H2O vapor absorption cooling system.
2. Steam Turbine Heat recovery using solar integrated double bed activated carbon-methanol and activated carbon-R134a Vapor adsorption refrigeration system for space cooling purpose.
3. Reheating Rankine power generation heat recovery of the condenser by using a solar integrated organic Rankine cycle for combined cooling, heating, and power generation effect.
4. Combined reheating and regeneration steam power cycle, analysis and process heat recovery through a vapor generator by using the Vapour jet refrigeration system.

All the above four systems are suitable for the low, medium, and high-grade temperature sources of heat recovery and produce the combined effect of energy efficiently.

A parametric study has been carried out to analyze some influenced parameters such as condenser temperature, turbines (GT&ST) output, ORC performance, cooling effect of VARS, and vapor adsorption refrigeration and ejector cooling system. The influencing effect of gas turbine inlet temperature, compression ratio with different combustion of natural gases provides the best performance of the GT system for operation of ST plant under the different operating conditions of the compressor, GT, and combustion chamber. This case concluded as the maximum exergy loss found in the combustion chamber of GT system and exhaust flow system of ST system in terms of 41% and 8%, respectively. The combined and exergetic efficiency of the plant is estimated to be 41% and 38.5% respectively. In the present statistical model, 4 levels and 3 factors (Pressure ratio, operating temperature and type of fuel gases) have been considered. Furthermore, overall efficiency, gas turbine efficiency, heat loss in GT plant, Exergy destruction in thermal utilities like Compressor, combustion chamber and gas turbine are investigated. The statistical modeling concluded that the comparative results of actual and predicted results at different compression ratio of combustible gases which affects the overall performance of combined GT-ST plant. This study helps to justify possible efficiency improvement by identifying the irreversibility of plant utilities.

The combined reheating-regeneration power generation analysis concluded that the energy- exergy analysis for the Boiler, turbines, Feed heaters, condenser and pump majorly. The result of the thermodynamic analysis is computed as 42% of plant thermal efficiency, 70 % of steam generation unit efficiency. Maximum heat absorbed by economizer of plant as 39% is achieved, and quality of steam was found around 89-90% with 40TPH of coal consumption. Boilers, HPT, IPT, Super heaters have found best performance in analysis and Reheating-Regenerative Rankine method improves 6-8% in thermal efficiency. It has been observed that energy efficiency (theoretical) is always more than energy efficiency (actual), which means it helps to understand the performance of thermal power plants and justify possible efficiency improvement with efficient power generation opportunities like waste heat recovery technology employment.

The performance of cooling systems of proposed research work is carried out the source temperatures available for both beds of Vapor adsorption refrigeration systems (VAdRS) from condenser exhaust, ETC solar system. The adsorbent and adsorbate pair for double bed VAdRS has been recommended by activated carbon as adsorbent and methanol and R134a as the adsorbate. The significant findings of present work are the maximum irreversibility found in Boiler as 47% in thermal power plants and solar generators as 12% of adsorption machines, whereas overall cooling effect from adsorption systems increases by 15% in double bed combination. EES software is used for all analyses. The VAM machine for stack flow heat recovery is performed by 0.708 of COP, a suitable cooling for space and water chilling purposes.

CHAPTER 1

Chapter-1 Introduction

1.1 Introduction

The proposed work of entire research has been performed an experimental implementation of the combine heating-power and cooling cycle with and without solar integrated. The present research work focuses on two major thermodynamic analyses: the integration of advanced thermodynamic cycles and the heat recovery system with employment of solar thermal systems and the second is complete thermodynamic analysis, consisting of Energy and Exergy analysis.

The present chapter explains the complete work conduct of proposed research, thermodynamic analysis, thermodynamic systems for heat recovery purposes, technological development status at the global and country level, solar thermal systems, proposed thermal model, and research methodology.

1.2 Thermodynamic energy-exergy analysis

Quantitative evaluation of energy of process or cycle can be done by FLT; it permits the quantity of energy evaluation only; the supply energy converts into two products (work) and waste (discharge energy). The actual performance of system is dealt by the 2nd law of thermodynamics or entropy generation concept. The entropy generation concept has been used for proposed research work. Complete Energy-Exergy analysis is applied for thermal modeling of all research models. Energy and Exergy equations for thermal utilities identify the cause of losses, locations, magnitudes of process inefficiencies, thermal in-equilibrium and sizing of thermal utilities.

1.3 Concept of combined cooling heating and power generation (figure 1)

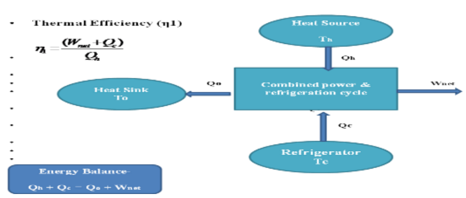


Figure 1: Thermodynamic concepts of power cooling system

1.3.1 Cogeneration system

Cogeneration or combined heat and power (CHP) is the use of a heat engine or power station to generate electricity and useful heat at the same time. Cogeneration is a thermodynamically efficient use of fuel. In separate electricity production, some energy must be discarded as waste heat, but in cogeneration, this thermal energy is put to use. All thermal power plants emit heat during electricity, which can be released into the natural environment through cooling towers, flue gas, or by other means. By-product heat at moderate temperatures (100–180 °C, 212–356 °F) can also be used in absorption refrigerators for cooling. Requirements of cogeneration may be met in many ways ranging from steam and gas turbines to fuel cells and Sterling engines.

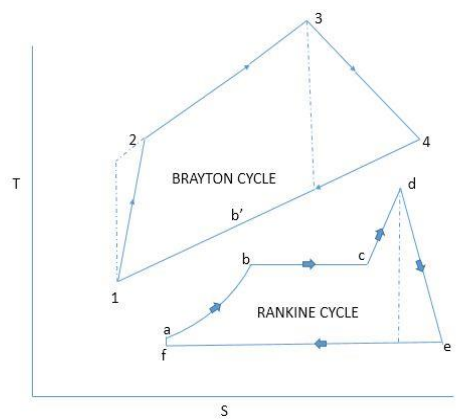


Figure 2: Combined power cycle [20]

The thermodynamic cycle of the basic combined cycle consists of two power plant cycles shown in figure-2 of the combined power cycle.

Topping and Bottoming Cycle

In a bottoming cycle, the primary fuel produces high temperature thermal energy and the heat rejected from the process is used to generate power through a recovery Boiler and a turbine generator. Bottoming cycles are suitable for manufacturing processes that require heat at high temperatures in furnaces and kilns, and reject heat at significantly high temperatures. Typical areas of application include cement, steel, ceramic, gas and petrochemical industries. Bottoming cycle plants are much less common than topping cycle plants. The waste gases coming out of the furnace is utilized in a Boiler to generate steam, which drives the turbine to produce electricity.

1.3.3 Trigeneration

Tri-generation technology can simultaneously provide three forms of output energy; electrical power, heating and cooling. Trigeneration is also known as CCHP (Combined Cooling, Heating and Power) or CHRP (Combined Heating, Refrigeration and Power) as shown in figure 3.

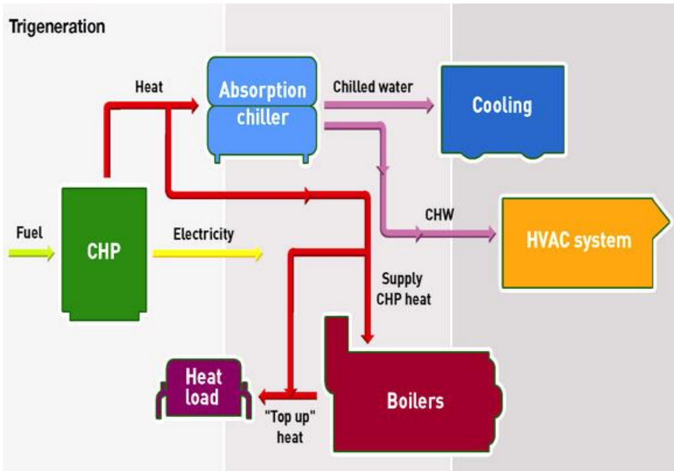


Figure 3: Trigeneration system layout [20]

1.4 Energy recovery system

(a) Organic Rankine cycle (ORC)

- The ORC applies the principle of the steam Rankine cycle, but uses organic working fluid with low boiling points, instead of steam, to recover heat from a lower temperature source. The cycle consists of an expansion turbine, condenser, a pump, a Boiler and a superheated (provide superheat is needed).[21]

ORC application-

- Different forms of combined with ORC as a bottoming cycle, ORC with different working fluid for power plants, cement industry, desalination, process industry, furniture manufacturing industry. [21]
- Wastage heat recovery

ORC working fluids-

- The working fluid of an organic Rankine cycle is very important. Pure working fluids such as HCFC123 (CHCl2CF3), PF5050 (CF3(CF2)3CF3), HFC-245fa (CH3CH2CHF2), HFC-245ca (CF3CHFCH2F), isobutene ((CH3)2C=CH2), n-pentane and aromatic hydrocarbons, have been studied for organic Rankine cycles. The slope of the saturation curve (ds/dt slope) of a working fluid in a Temperature-Entropy diagram .Working fluids are classified as a dry fluid, wet fluid or isentropic fluid, it is defined by. If J=ds/dt, J.>0 for dry fluid (pentane),J=0 for isentropic fluid (R11) and J<0 for wet fluid (water) [21].Isentropic or dry fluid was suggested for ORC to avoid liquid droplet impingement in turbine blade during the expansion. If the fluid is too dry the expanded vapor will leave the turbine with substantial superheat, which is a waste and adds to the condenser's cooling load in the condenser. The cycle efficiency can be increased using this superheat to preheat the liquid after it leaves the feed pump and before it enters the Boilers. All components of ORC shown in figure 4 ORC components [21].

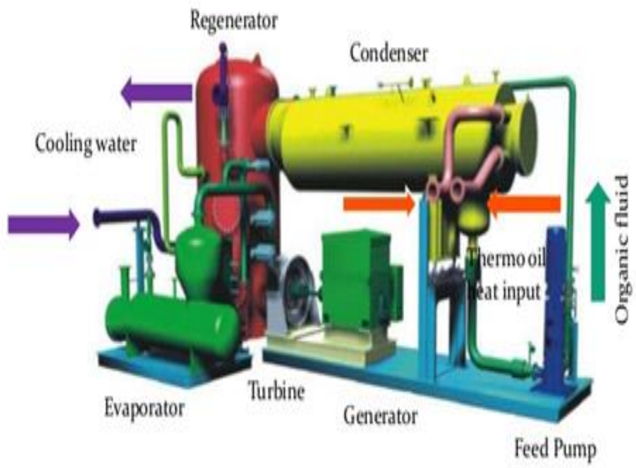


Figure 4: ORC components [21]

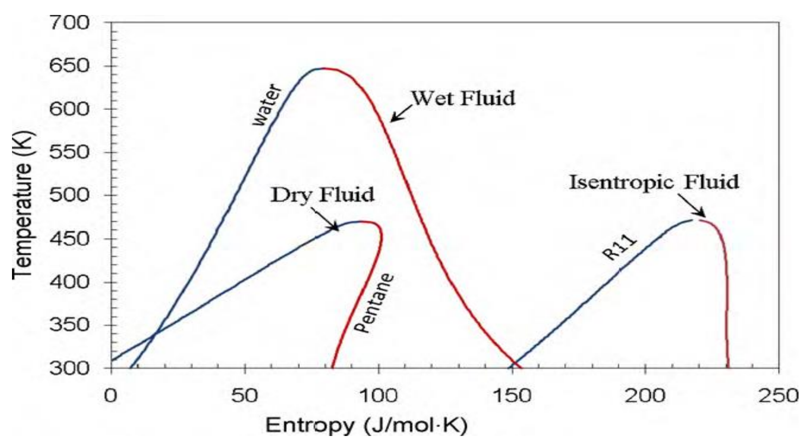


Figure 5: Temperature-Entropy diagram of fluids [21]

Required working fluid properties-

The main characteristic is that the working fluid's boiling occurs over a range of temperatures (figure 5).

Binary fluid (NH3-H2O) thermal System[21]

The Binary fluid (NH3-H2O) Thermal System (figure 6) was 1st developed by Alexander Kalina in the late 1970 and early 1980 for generating electricity using low to medium temperature recourses. Kalina system uses ammonia-water (Binary fluid) mixture as working fluid. This mixture is used as source for power and cooling. In this cycle ammonia is the refrigerant and water are the absorbent due to the high difference in their boiling point and high enthalpy.

- In the Kalina system, used binary fluid mixture results in a good thermal match in the Boiler due to the non-isothermal boiling created by the shifting mixture composition. Several studies have shown that the Kalina cycle performs substantially better than a steam Rankine cycle system. A second law analysis showed that by using a binary fluid, the Kalina cycle reduced irreversibility in the Boiler, resulting in improved efficiency.
- One drawback of the Kalina system is that a high vapor fraction is needed in the Boiler; however, the heat exchanger surface is easy to dry out at high vapor fractions, resulting in lower overall heat transfer coefficients and a larger heat exchange area. Another drawback relates to the corrosively of ammonia. Impurities in liquid ammonia such as air or carbon dioxide can cause stress corrosion cracking of mild steel and also ammonia is highly corrosive towards copper and zinc.

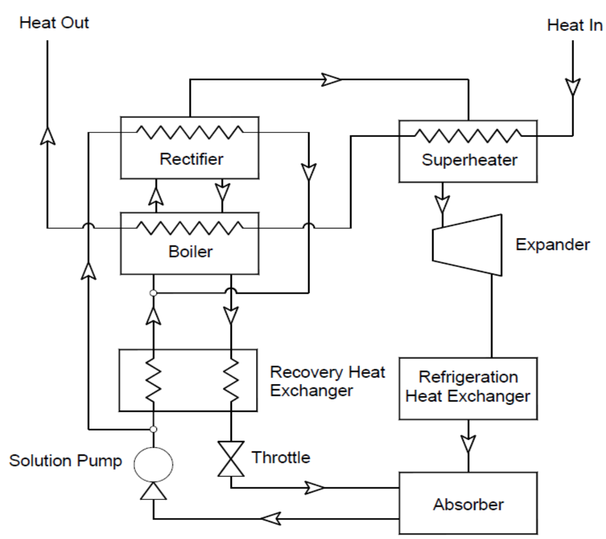


Figure 6: Binary fluid (NH3-H2O) -Thermal System [21]

(b) Combined Rankine and Refrigeration cooling cycle

- Goswami [21] proposed novel thermodynamic cycle that uses binary mixture to produce power and refrigeration simultaneously in one loop. This cycle is a combination of Rankine power and absorption cooling cycle (figure 7). Its advantages include the production of power and cooling in the same cycle, the design flexibility to produce any combination of power and refrigeration, the efficient conversion of moderate temperature heat source and the possibility of improved resource utilization compared to separate power and cooling system. The binary mixture first used was ammonia-water and later on new binary fluids were proposed and studied.

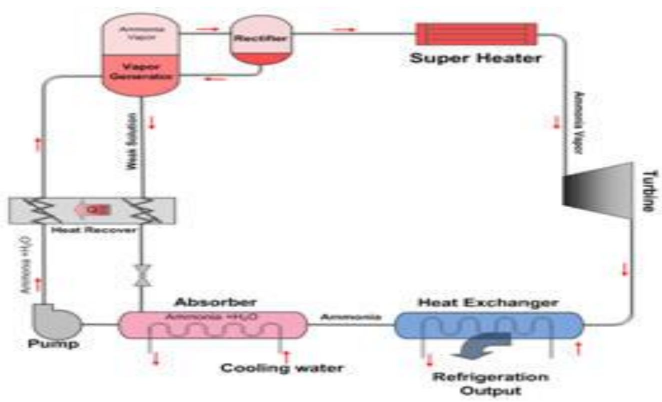


Figure 7: Combined Rankine and Refrigeration cooling cycle-[21]

(C) Vapour jet refrigeration (VJR) Techniques

The vapour jet refrigeration system is used to entrain low pressure fluid by high pressure fluid without mechanical power input (figure 8).

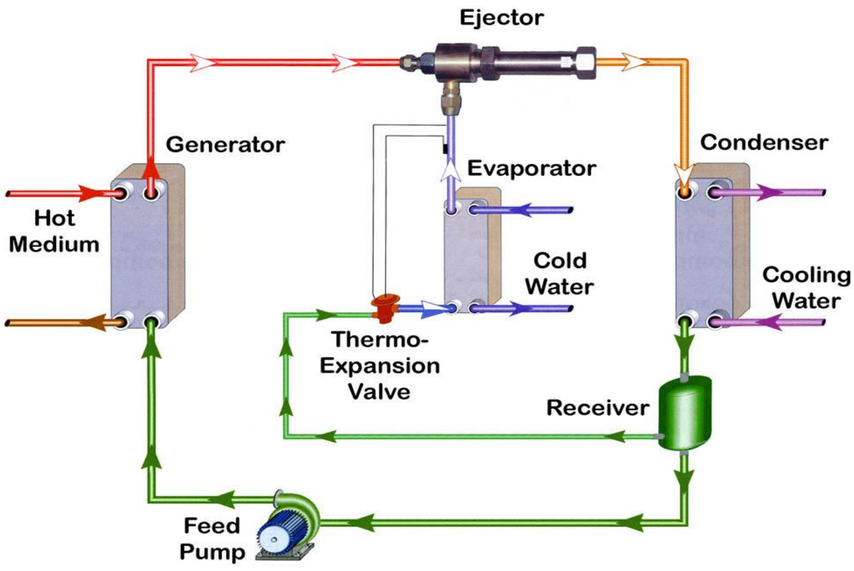


Figure 8: Vapour jet refrigeration cooling cycle[21]

1.5 Energy recovery (High-Low grade) technology

Three categories of wastage heat sources are distinguished concerning the temperature level, low (<2300C), medium (230-650 0C) and high (>6500C).Waste heat sites and thermal levels are listed in table 1 of energy recovery and its sources.

Table 1: Energy recovery and its sources

Heat categories	Heat sources	Temperature in 0C	Suggested recovery technology
High grade (>6500C)	Solid waste	650-1000	Air preheating
	Nickel refining furnace	1370-1650	Steam generation for heating
	Copper reverberator furnace	760-815	Thermoelectric and thermal PV
	Glass melting furnace	1000-1550	Heat exchanger for preheating
	Hydrogen plant	650-1000	Thermal PV

Medium grade (230-6500C)	Steam Boiler exhaust	230-480	Steam Rankine cycle
	Gas turbine exhaust	370-540	Organic Rankine cycle
	Drying and baking ovens	230-600	Thermal PV
	Catalytic crackers	425-650	Thermal PV
	Reciprocating engine exhaust	315-600	Thermoelectric
Low grade (>2300C)	Welding and injection molding	32-88	Kalina cycle
	Hot processed liquids and solids	32-233	Organic Rankine cycle
	Drying, Baking and Curing ovens	93-230	Absorption and adsorption cooling
	Bearing	32-88	Piezoelectric

1.5.1 Status of energy recovery technology in India

- Cement industry

According to a study done by Turboden (one of the market leaders in ORC technology), a 2,500 ton per day plant of cement can set up a 1.6 MW. Waste heat recovery plant using the Organic Rankine cycle. Based on this assumption and the cement industry's projected manufacturing capacity in India, a rough potential of the electricity production from waste heat by ORC technology is estimated to be 574 MW.

- Glass industry

Typically, a 500 ton per day glass manufacturing plant will have potential for a 1 MW Organic Rankine recovery plant. The projected waste heat recovery potential through ORC in India's glass industry is total potential estimated at around 36 MW by 2017.

- Iron and steel industry

Typically, a 6,000 ton per day steel rolling mill can generate 2.4 MW of electricity through an Organic Rankine recovery plant. The projected waste heat recovery potential through ORC in the Iron and Steel sector is total potential estimated to be around 148.4 MW by 2017.

1.5.2 Commercial WHR status in India

Table-2 shows the commercial capacity of heat recovery power generation systems for industry with the application of renewable energy sources in India using ORC.

ORC status in India

Table 2: ORC development in India [127]

Sector	Capacity (MW)
Wastage Heat Recovery in Major Industry	
Cement	574.2
Glass	35.7
c. Iron and steel	148.4
Total	758.3
2.Renewable energy	
Solar thermal energy storage	1440
Biogas plant	2208
Geothermal	NA

Total	3648
Grand Total	4406.3

1.6 Thermodynamic models for research work

1. Stack flow heat recovery of combined GT-ST plant using LiBr-H2O vapor absorption cooling system.

2. Steam turbine heat recovery using solar integrated double bed activated carbon-methanol and activated carbon-R134a vapor adsorption refrigeration system for space cooling purpose.

3. Condenser heat recovery of reheating steam turbine system by using solar integrated organic Rankine cycle for combined cooling, heating and power generation effect.

4. Regeneration-reheating Rankine power generation with eco-friendly refrigerant based vapour jet refrigeration for heat recovery

1.7 Research work methodology

Two methods of research analysis have analyzed the proposed thermodynamic heat recovery models, one method is adopted as complete thermodynamic (energy-exergy) modeling of discussed heat recovery systems for actual performance evaluation and losses identification whereas second method is implemented as design of experiments (DOE) based statistical analysis of performance parameters for actual & predicted results comparison, and system optimization as per statistical results. The statistical based mathematical analysis like multi-linear regression (MLR) and Taguchi methods have been used in present research.

1.8 Organization of thesis

This thesis is structured with mainly introduction, literature review, system study thermodynamic analysis & methodology, and result discussion. The literature review systematically reviews the research efforts made on combined GT-ST cycle; various heat recoveries cooling systems like vapour absorption, adsorption and ejector refrigeration system, heat recovery by organic Rankine system.

The thesis work is divided into 6 chapters of research work including research publication. The chapters of thesis are distributed below further-

Chapter-1 Introduction describes the brief of entire work of research conducted during the PhD.

Chapter-2 Literature review depicts the thoroughly research reviews in different aspects of all mentioned systems, used analysis, research, and commercialized system applications. The research gap has been discussed in same chapter as per present literature.

Chapter-3 Thermodynamic systems chapter configured all four proposed heat recovery thermal systems with its operation and complete plan schematic.

Chapter-4 Thermodynamic (energy-exergy) analysis of thermal systems, evaluate all systems by using energy conservation and entropy generation methods. The mathematical soft computing techniques also apply for comparative analysis of experimental and predicted results.

Chapter-5 Results and Discussion deals the all parametric results with possible performance of all thermodynamic systems.

Chapter-6 Conclusion and future scope, summarizes overall conclusions of the result obtained in the above study and a little effort will be made to search out a domain for further research in the proposed area of interest.

Publications

CHAPTER 2

Chapter-2 Literature review

2.1 Introduction

Energy-Exergy analysis of thermal systems provides the opportunity to improve and identify the location of losses and losses mainly occurs during the operation and it remarks as exergy destruction.

An extensive review of the literature has been done on different thermal systems arrangement with idea of cogeneration and tri-generation effect utilizing waste heat from power plants in the present chapter. The main idea was to have possible future direction of research. The literature review has been classified as under:-

1. Concept of Solar integrated Cogeneration and Tri-Generation thermodynamic systems.
2. Idea of Organic Rankine Cycle with different working fluids for heat recovery purpose.
3. Thermodynamic modeling of WHR systems using energy-exergy-emergy and entropy generation principle.
4. Feasible Technological development and commercial application of WHR systems in global level.
5. Environmental aspect of proposed WHR systems.

2.2 Literature survey

The research review focuses on the detail energy-exergy analysis of all kinds of thermodynamic systems that produce the effect of combined heating power and cooling effect.

Amano et al. [1] proposed single effect aqua-ammonia refrigeration with power generation system. The proposed model is referring to kalian model of combined cooling & power system for low temperature application.

Xuet al. [2] carried out the parametric analysis of aqua-ammonia based vars with Rankine power generation system; he found that the efficiency of boiler, generator is influencing factors.

Angelino et al. and Kalina [3,4] highlighted that kalian model development and discuss the its advantages, and heat recovery scope in future.

Vijaya Raghavan and Goswami [5] investigated that the performance of kalian model of combined cooling-power and concluded that kalian model with ORC is suitable for heat recovery distillation purpose.

Vidal et al. [6] carried out exergy evaluation for combined cooling and power cycle. The aqua- ammonia cooling system was analyzed through ASPEN plus.

Zheng, Oliveira et al.[7,8] presented the novel approach of absorption cooling and power cycle with extensive work fluid properties review. The ORC system has been analyzed for different operating conditions of use refrigerants.

Padilla et al. [9] highlighted that the performance analysis of aqua-ammonia system with orc performance and found the effect of absorber, generator temperature on plant's overall performance.

Pouraghaie et al.[10] assessed thermodynamic performance optimization of a combined power and cooling cycle.

Wang, Chacartegui et al [11,12] investigate that the eco-friendly based orc for combined bottoming plant heat recovery.

Liu et al [13] carried out that thermals heat recovery systems for small scale plant of HVAC, this system can be applied for the range of cooling effect of window AC systems.

Wang et al [14]have developed the small size of ORc system which has 5 KW and 0.8 of COP. They carry through the microchannel for heat transfer. The system can be used to recover and convert waste heat directly into cooling.

Srinivas [15] has summarized the different thermal utilities like recuperator, heat exchangers, condensers for the heat recovery.

Kontomaris [16] has reviewed ORC cycle efficiency with different operating conditions and estimated thermal efficiency using many organic working fluids.

Hung et al and Hou et al [17,18] presented the overview of chemical-fluids for heat recovery purpose using different-different thermal systems. The slope between temperatures and Entropy for working fluids shows the dry and wet fluid behavior for ORC systems, and experimented the ORC for the internal combustion engine and CCHP plant operation.

Quoilin S et al [19]reviewed the selection and properties of the working fluid in ORC, the working fluid must have a low value of ODP&GWP as environmental concern and also full fill the several criteria, such as being economical, non-toxic, non-flammable, environmentally friendly, etc.

Maidment and Tozer [20] have reviewed a number of combined energy production plants operating in global markets. They analyzed different schemes of combined energy production including different cooling and engine technologies. Cogeneration is a thermodynamically efficient technology for power and energy in terms of electricity generation, some energy must be discarded as waste heat, but in cogeneration this thermal energy is put to use. All thermal power plants emit heat during electricity, which can be released into the natural environment through cooling towers, flue gas, or by other means. By-product heat at moderate temperatures (100–180 °C, 212–356 °F) can also be used in absorption refrigerators for cooling. Requirements of cogeneration may be met in many ways ranging from steam and gas turbines.

Riffat & Erickson et al. [21,22] carried out the experimental performance of ORC. The VARS also plays important role in the combined cooling effect with the power cycle. The estimated COP was found 0.34 which is a considerable range.

Maizza et al [23] summarized heat-transfer properties of the working fluids are of great importance. Desirable properties are: low viscosity and surface tension, the low specific heat of the liquid, high thermal conductivity, high latent heat of vaporization. The evaporator enthalpy ratio i.e. the ratio of enthalpy of vaporization to the sensible enthalpy required to raise the temperature of the compressed liquid should have a high value.

Ghodsipour and Sadrameli [24-26] conducted an experimental study on a rotary regenerator by solving a developed mathematical model and optimized with the experimental design method. In this way, the effect of dimensionless parameters on the effectiveness of rotary heat exchangers was investigated.

Khaliq and Kaushik [28, 30] conducted a second law analysis for the performance evaluation of gas turbine cogeneration system with reheat and defined the energetic and exergetic efficiency.

Also computed exergy destruction in each component of above system.

B.F.Tchanche et al [29] studied about Absorption/Adsorption, cooling, heating, heat to electricity conversion, from waste heat can be achieved using thermodynamic power cycles, Rankine, Organic Rankine (ORC), Kalina, Goswami, and Trans-critical cycles and its high, medium ,low temperature recourses based industrial machines or systems explained in this paper. Solar ORC reverse osmosis desalination system is still at the research stage.

Omendra Singh,et al [33] computed the thermodynamics of Trigeneration (Brayton-Rankine- Kalina combined power cycle) power (TRIGEN) system explained in this paper. The main focus of methodologies is on thermo economic analysis of Trigen.

A Khaliq et al [34, 35] carried out the energy & exergy analysis of ejector-absorption system, and found 8 % of exergetic efficiency. The proposed system is applicable for the system combination of gas turbine and absorption chilling machine.

Tiwari and Karimi [36, 37] reviewed the different technology for low grade energy recovery system for the process industry with suitable working fluid properties and also explained the brief of future refrigerants, another research concluded by authors on energy and exergy efficiency of different commercial fluid for wastage energy recovery system and introduce theoretical thermal performance of new trend of refrigerants like R1234yf.

Dincer and Rosen [31,32, 38] highlighted the importance of energy-exergy principle on various thermodynamic system, and find out the losses of each components.

A.Bejan [39] introduced EGM (Entropy generation minimization) method, and concluded whenever entropy production is minimized, useful energy is maximized.

Khaliq and Kaushik [40] carried out the exergy analysis of different parts of GT plant and found that the 50% of combustion efficiency. Also computed the performance of combined GT-ST power generation system by considering the effect of intermediate pressure-ratio..

Yang et al [41] compared the conventional and advanced approach of exergy analysis of supercritical coal thermal power plants. Conventional approach identified exergy destruction of all components, whereas advanced exergy analyses conclude thermodynamic interactions among thermal utilities for energy saving potential.

Ahmadi and Dincer [42] carried out the energy-exergy of dual pressure combined cycle power plant (CCPP),and found the exergy destruction of duct burner of HRSG, but the power output of CCPP is improved.

S.C.Kaushik et al and Aljundi [43,44] carried out energy-exergy analysis using entropy generation principle and find out max exergy destruction in the boiler in case of st power plant and cc in the gt power plant.

Gulhane and Thakur [45] investigated the amount and source of irreversibility in 35 TPH capacity of Boiler. Furthermore, found 76% of exergy destruction at peak load on Boiler but 1st and 2nd law efficiency are enhanced.

Osueke et al [46] computed maximum exergy destruction 87.3% in Boiler, and plant exergy efficiency 11.03% with the effect of water flow in Boiler on the steam generation and plant output of 75MW of steam power plant, which is located in Sapele-Nigeria.

R Kumar [47] critically summarized the exergonomic(4E) analysis of the combined GT-ST power generation system, and found maximum exergy destruction in Boiler and condenser of st power plant , also found out ex dest in the combustion chamber of a gas-turbine power plant.

Vélez, F,B.Saleh and Quolin, S [48-51] showed the utilization of dumped heat (waste heat) into the atmosphere from power plants, industry like glass, cement, textile, etc. by the method of heat recovery coupled with Organic Rankine Cycle (ORC) based renewable energy conversion system for combined effect of cooling-heating and power. The ORC applies the principle of the steam Rankine cycle but uses an organic working fluid, with a low boiling point, instead of steam to recover heat from a lower temperature source. ORC is a capable technology for the conversion of these low temperature sources to power.

Chen H,et al [52,53-65] has studied the working fluid for the ORC system and its thermal property, selection criteria, they have been found isentropic type fluid R 134a is applied in ORC system. In ORC, superheating is unnecessary while applying isentropic fluids because its expansion would not cause any wet fluid problem. Some fluids likeR-141b, R-123, R-21, R-245ca, R- 245fa, R-236ea and R-142b etc., have critical temperature above 400 K, resulting from being used only in the supercritical cycle for low grade heat sources. R134a is belonged from Group-A1 of ASHARE, that means nontoxic and non-flammable fluid with zero Ozone depletion potential (ODP) value. R134a have wide variety of turbo-machines, heat exchangers.

Franco, A and Vaja I. [66-69] have conducted a simulation-based study of two stage ORC available for desalination of an upper ORC coupled with a lower ORC with R134a as the working fluid obtained an efficiency of 4.2% and a global efficiency of 3%. Furthermore, other studies of ORC with R134a as the working fluid for applications in geothermal sources and in bottoming cycles with internal combustion engine have been reported.

Saleh B. and Tchange B.F. [50, 70] studied 31 pure component working fluids for Organic Rankine cycle has shown an efficiency of 7.7% being achieved in cycles that operate with R134a at temperature of 100.0ºC and other fluids were studied and it was reported that R134a is the most suitable. The thermodynamic modeling of ORC based micro CHP discussed efficiency, water flow rate and achievable electricity production rate.

Chen et al [71-74] Thermodynamic evaluation of basic ORC with arrangement of internal heat exchangers, examined at 11-14% of maximum efficiency at conditions of saturation & overheating of working fluid, and also concludes the effect of pressure ratio and overheating of fluid in performance of system. Other works that have analyzed by the researchers use of R134a as working fluid in ORC for low grade heat recovery. There is a great interest in the use of this fluid for generation of energy in systems with heat input temperature below 150ºC. However, we noticed in the literature about the best thermodynamic conditions for its use and a gradual impact has been observed working with

R134a fluid at superheated conditions. In the influence of the source of energy on the performance of the system as well as on the net specific work output derived from the system.

BaijuV.and Murleedharan C [75] studied the single of multiple bed of the adsorption refrigeration system. And found that output cooling is not continuous in a single adsorption refrigeration cycle system. A minimum of two adsorber is required to obtain a continuous cooling effect.

Anyanwu EE reported [76, 77, 78] reported that the solar based adsorption refrigeration system

Saha et.al [79] have examined that two stage non-regenerative adsorption chiller prototype with experimental results show that the system operated with temperature range of 55-30 0C.

Dincer and Rosen [80,81-83] explained that exergy importance and its requirement for thermodynamic system development.is the maximum rate of work, which is obtained from it and passes reversibly to the environmental state. Exergy analysis is the theoretical limitations upon a system, clearly pointing out that no real system can conserve exergy and that only a portion of the input exergy can be recovered.

A.Bejan [84] introduces entropy generation minimization mechanism, if entropy production is minimized, then useful energy is maximized.

Khaliq and Kaushik [85] concluded the combustion chamber exergy destruction is 50% of overall exergy destruction of the cycle, and effect of intermediate pressure-ratio and effect two stage of reheating on combined GT-ST power cycle performance by using 2nd law thermodynamic approach.

Yang et al [86] compared conventional and advanced approach of exergy analysis of supercritical coal thermal power plant. Conventional approach identified exergy destruction of all components, whereas advanced exergy analyses conclude thermodynamic interactions among thermal utilities for energy saving potential.

Kaushalendra and Mishra [87-91] found maximum exergy destruction in Boiler part of thermal power plant analysis. Compressor of conventional refrigeration consumes larger amount of electrical energy with high destruction rate in VCRS, and author also summarize different thermodynamic systems for low to high grade energy recovery from industrial & power plant sector for combined heating-cooling and power generation by using new trends of working fluid in advanced thermal cycles like organic Rankine cycle, kalian cycles supercritical organic Rankine cycle etc. in efficient manner with environmental aspect, and also reviewed status of industrial waste heat recovery in order of more than 45% of energy generation supply to industrial purpose. The industrial sector (oil refineries, cement, glass, steel etc.) and power plant discard large amount of heat with useful energy generation or energy in production into the atmosphere. The novel concepts of wastage heat (low and high grade) recovery technology like ORC, Tri-generation, Co-generation,HRSG, Vapor absorption and adsorption refrigeration system with and without integration of solar energy utilization recover waste heat from several process like, kiln and clinker gas flow recovery in cement production plant, oven exhaust heat recovery in glass industry, pre-heating exhaust heat recovery of rolling process in steel industry, and produce power, heating and as well as cooling generation.

Ghaebi et al. [92] have carried out the energetic-exergetic analysis of gas turbine system for trigeneration effect and CCHP.

Balli et al. [95] examined the thermo economic analyses of a trigeneration (TRIGEN) system with a gas diesel engine and found that 6.5 MW of power output from diesel set.

Al-Sulaiman et al [96-97,98] carried out orc based trigen system and found the exergy analysis of different parts of trigeneration unit also found the performance of solar PTC system.

Deng et al. [93-95] highlighted that exergy analysis of micro-trigen system and performance of cchp unit, another study also carried out for energine heat recovery.

Kong et al. [99-103] studied the linear programming model which gives the results of optimum energy combination for the CCHP system with a gas turbine plant. Another study about the energy management and optimal operation of a micro-combined cooling, heating and power system

Bandyapadhyay et al [104-107] examined the optimization of cogeneration plant with working fluid availability. Another study of thermo economic analysis of the cogeneration plants. Air and combustion products were regarded as ideal gases for this study and the isentropic performance have consider for the utility's performance. The entropy generation concept used for minimize the exergy production costs

Gomri and Hakimi [108, 109] carried out exergy analysis of double effect Lithium Bromide- water absorption refrigeration system. Also concluded that the highest exergy loss occurs in the absorber and in the HP generator and therefore the absorber and HP generator are the most important components of the double effect refrigeration system.

Kaushik and Arora [110] carried out the energy and exergy analysis of libr-h20 vars and found that maximum irrerversibity in the absorber. Also found the effect of generator temperature ,on exergy performance.

Khaliq et al [111] carried out complete thermodynamic analysis of waste heat based combined power and ejector-absorption refrigeration cycle. And fund 53.6% of the total input exergy is destroyed due to irreversibility in the components, 22.7% is available as a useful exergy output, and 23.7% is exhaust exergy lost to the environment.

Thamir K. Ibrahim et al [112] summarized the performance assessment of cchp plant using entropy generation concept, and he suggested that reducing the exergy loss in hrsg of cchp plant.

[113-123] Carried out the pant performance using several statistical model Another type of plant data analysis is deals by statistical modeling techniques like Regression model, least square method, Maximum likelihood estimator (MLE) method, Auto regression integrated moving average model (ARIMA),Multiple linear regression (MLR). Artificial neural network (ANN), Response surface method (RMS), Multivariate regression (MVR), etc. for error identification, parametric comparison and complex problem solving.

The author Carried out the orc based geothermal plant analysis using statistician analysis tool for parametric optimization and system design. The parametric analysis of geothermal power plant is estimated by using multiple linear regression method, and ANOVA-I&II mathematical tools have been conducted for organic Rankine cycle performance analysis and compared with R-analysis tool for error estimation and parametric comparison in recent research work [124-125].

Kaushalendra and Mishra [126] estimated parametric comparison of combined reheating-regeneration thermal cycle by using Taguchi method of statistical method and concluded that the turbine inlet temperature is main factor for overall efficiency increment.
Jing Li et al [113] studied on novel system of turbine-driven steam organic Rankine cycle which is used in DSG (Direct Steam Generator) system. The thermodynamic performance of this innovative system is investigated in terms of equivalent payback period (EPP), multiple steps of heat discharge, etc. The results of this analysis indicate that the second step ejected heat can enhance the storage capacity by 460%, with an EPP of less than 5 years in several cases. Overall, the proposed solution improves the cost-effectiveness of the DSG system.

2.3 Conclusions from literature review

The literature review revealed the following

- In general, conventional steam cycle operates in medium to high temperature and cannot be cost effective at smaller scale or low temperature resources. In The low medium temperature range Organic Rankine, Kalina, Goswami, and Trans-critical cycles have demonstrated.
- From the literature, it is evident that an integrated approach towards operational efficiency improvements of existing systems, reduction of losses in operational mechanisms, end-use efficiency and also making use of waste heat recovery technologies are very much essential for energy conservation.
- The organic Rankine cycle applications in waste heat recovery. Working fluid properties and selection (including pure fluids and mixtures) was reviewed. Also, some important physical properties of the working fluids for ORC and the

performance of the system were introduced. Different applications of ORC systems including solar thermal, biomass ORC, solar thermal reverse osmosis desalination, geothermal application, and waste heat recover from industrial process were intensively investigated. The paper also presented the different employed expander in the ORC system and introduced many factors which should be considered such as the power capacity, isentropic efficiency, cost and complexity their application range. Environmental concern over climate change as well as energy price is reasons supporting application of the waste heat recover by the ORC technology.

- Reasonable focus has been paid on analysis and understanding of various aspects related to effectiveness of energy use. Still, in each area, potential exists for sizable research work.
- Different Waste Heat Recovery Systems (WHRS) have been experimented to analyze the effect of various parameters of energy savings.
- The life cycle assessment and Emergy Method analysis of heat recovery systems and Trigeneration Systems gives the comparative results of qualitative energy utilization, and system optimization information with the comparison of other power generation techniques like, wind, geothermal, solar, hydro power, etc.

2.4 Research gap and identification.

- The low-temperature ORC industry still awaits the arrival of a low GWP replacement fluid exiting fluid to maintain the cost advantage that can be realized by using equipment for ORC duty. The newly developed fluid 1233zd(E) could be that fluid.
- The medium pressure new refrigerants R1234yf and R1234ze(E) approach R134a and R245fa..
- Till now there is no study about solar based combined heating, power and cooling system analysis using HFO-1234yf as working substance in Organic Rankine cycle for food process industry in India.

- The above described energy status and its future demand indicates the research and development in area of energy efficient technological development, its utilization with proper management in cost effective and eco-friendly nature for commercial, domestic and industrial purpose. The technology of vapor adsorption refrigeration system has potential for recover heat of low and high grade both, as well as integrated with solar thermal energy utilization. Many researches have been conducted in past, but implementation of technology and its development is still awaited.
- Statistical and soft computing methods can help to identify the influencing factor for efficient plant operation. Lot of work is to be need for further improvement in area of parametric & system optimization, economics modeling of power generation plant.

CHAPTER 3

Chapter-3 Thermodynamic systems

The chapter's present title covers the detailed study of all proposed thermodynamic systems and its thermal analysis. The four thermal systems have planned heat recovery of the power plant and capacitive power consumption industry with heating-cooling and power generation. The thermodynamic analysis of systems deals with 1^{st} law and 2^{nd} law of thermodynamics (energy conversion & entropy generation); the mathematical optimization techniques have implemented in terms of Taguchi, multilinear regression method parametric optimization, comparative study. The all thermodynamic system has been projected systematically in this chapter.

Model-1 Condenser heat recovery of the reheating steam turbine system by using a solar integrated Organic Rankine cycle

A two-way flow control valve is provided at the air inlet of heat HE 2 to regulate the air's mass flow rate. Heat, which is rejected from the air, is added to the heat exchanger's superheated refrigerant vapour. The cold air obtained at the air outlet of HE 1 is sent for space cooling or chilling. The superheated refrigerant vapour obtained at the ORC outlet of HE 2, at a higher temperature, passes through HE 3. The mixture of hot air extracted from HE 1 and HE 4 is used as the hot fluid in HE 3. A three-way flow control valve is provided at the junction of the two extraction lines to regulate the mass flow rate of the hot air from HE 1 and HE 4 going into the mixture. Heat is rejected from the hot air and added to the superheated refrigerant vapour in the heat exchanger. The cooled air is released into the atmosphere from the outlet of HE 3. A bypass line is provided across the solar heat exchanger to bypass it in case of the solar heater's condition and the solar heat exchanger not being used. Another bypass line is provided bypassing expander 2 and HE 4 in case of the same. A three-way flow control valve is provided at the inlet of the solar heat exchanger bypass line to enable or disable solar heat exchanger bypass. Another three-way flow control valve is provided at the outlet of the solar heat exchanger bypass line, to prevent the backflow of the superheated refrigerant vapour coming out of the solar heat exchanger, into the bypass line of the solar heat exchanger, in case the solar heat exchanger is in use, or to prevent the flow of the superheated refrigerant vapour into the outlet line of the solar heat exchanger, in case the solar heat exchanger is not in use. A three-way flow control valve is provided at the inlet of the expander 2 - HE 4 bypass line to bypass these systems when the solar heat exchanger is not used. Another three-way flow control valve is provided at the outlet of the expander 2- HE 4 bypass line to prevent the flow of the superheated refrigerant vapour, coming out of HE 4, into the expander 2 - HE 4 bypass line, in case the solar heat exchanger is in use, or to prevent the backflow of the superheated refrigerant vapour into the outlet line of HE 4, in case the solar heat exchanger is not in use.

Case I: With Solar Operated

The superheated heated refrigerant vapour obtained at a higher temperature from the ORC outlet of heat exchanger 3 then passes through the solar heat exchanger. A two-way flow control valve is provided at the solar heat exchanger's water inlet to regulate the mass flow rate of solar heated water into the solar heat exchanger. Heat, which is rejected from the hot water, is added to the superheated refrigerant vapour in the solar heat exchanger. The superheated refrigerant vapour was obtained at a higher temperature. Heat is rejected from the refrigerant vapour and added to the air in the heat exchanger. The hot air obtained at the heat exchanger 4 is sent to fulfill industrial process heat requirement. The refrigerant vapour is received at the ORC outlet of heat exchanger 4 with a lower temperature. A tapping is provided from the air outlet of heat exchanger 4 for extracting hot air to reheat the refrigerant vapour in the heat exchanger 3 if required. A three-way flow control valves are provided at the tapping junction at the air outlet of heat exchanger 4 to regulate the mass flow rate of hot air being extracted. A two- way flow control valve is also provided at the outlet of this extraction line, to prevent the flow of hot air extracted from heat exchanger 1 into this extraction line when it is not being operated and also to regulate the mass flow rate of the hot air being extracted from the outlet of the heat exchanger 4. The refrigerant vapour obtained at the heat exchanger 4 is at a lower temperature and enters the condenser heat exchanger. Any makeup of refrigerant, if required, is provided from the refrigerant tank.

Case II: Without solar operated system

The superheated heated refrigerant vapour obtained at a higher temperature from the ORC outlet of heat exchanger 3 then passes through the solar heat exchanger bypass line followed by expander 2- HE 4 bypass line and is fed into the condenser heat exchanger. Any makeup of refrigerant, if required, is provided from the refrigerant tank in between HE 4 outlet and condenser heat exchanger inlet.

Schematic (Figure 9) and description (table 3) of the proposed model for combined cooling, heating, and power generation by condenser heat recovery has been made using the software AUTOCAD 2016.

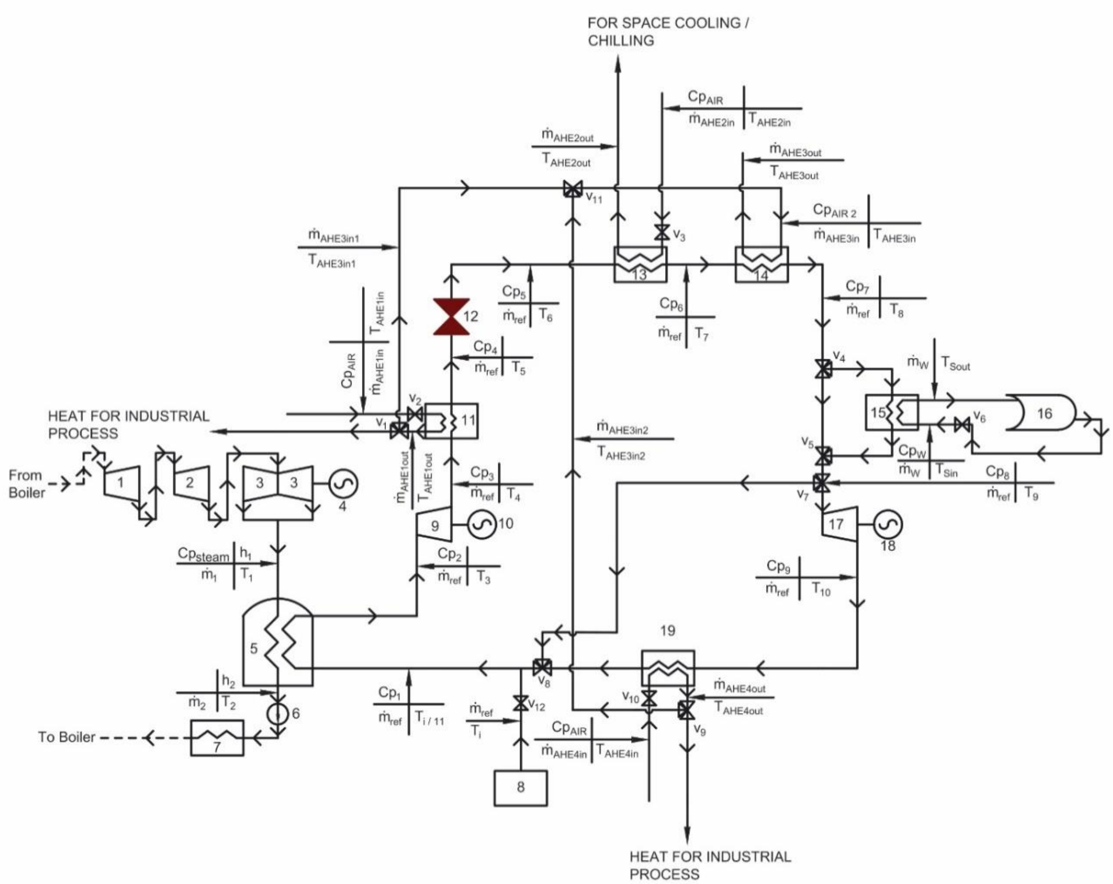


Figure 9: Schematic of combined cooling, heating, and power generation plant using condenser heat recovery

Table 3: Description of systems used in the schematic (figure:3)

System number	System description
1	High pressure turbine
2	Intermediate pressure turbine
3	Low pressure turbine
4	Power generator
5	Condenser
6	Circulation Extraction pump
7	Feedwater Heater
8	Refrigerant tank
9	Expander 1
10	Power generator
11	Heat exchanger 1
12	Throttle
13	Heat exchanger 2
14	Heat exchanger 3
15	Solar heat exchanger
16	Solar heater
17	Expander 2

18	Power generator
19	Heat exchanger 4
V1	Three-way flow control valve
V2	Two-way flow control valve
V3	Two-way flow control valve
V4	Three-way flow control valve
V5	Three-way flow control valve
V6	Two-way flow control valve
V7	Three-way flow control valve
V8	Three-way flow control valve
V9	Three-way flow control valve
V10	Two-way flow control valve
V11	Three-way flow control valve
V12	Two-way flow control valve

Model-2 Steam turbine heat recovery using solar-integrated double-bed activated carbon-methanol and activated carbon-R134a vapor adsorption refrigeration system for space cooling purposes.

System descriptions and operation

The proposed Rankine cycle was directly adopted from Kaushalendra and Mishra exergy-energy analysis of 9 MW thermal power plants [27]. The provision of double bed vapor adsorption system with Rankine model in this paper consisted of two bed of vapor adsorption, first bed of VADRS filled by activated carbon-methanol type adsorbent-adsorbate (refrigerant) pair, which recover exhaust heat of condenser of unit and produce cooling effect, another bed of VADRS occupied with activated carbon-R134a pair of adsorbent-adsorbate (refrigerant) and derived by solar water heater system. The selection of above pairs basically depends on source temperature for adsorption bed and thermo physical properties of pair for required cooling load. The all components are shown in figure 10 and table-4-unit description. The main components of VADRS are generator for heating of bed (generators may be solar thermal system or any thermal utilities), adsorption bed, condenser, evaporator and chilling chamber. Stainless steel fabricated adsorption bed consisted of two chambers, inner chamber (9& 17), which is filled with activated carbon as adsorbent and methanol for first VADRS &activated carbon-R134a for second VADRS as a adsorbate or refrigerant for adsorption cooling, and outer chamber (8& 16) for hot water chamber for desorption process of pair used, that hot water is supplied from condenser of unit to bed-1, and similar from ETC solar thermal collector (24) to bed-2.In the inner chamber, one end was connected to the condenser (11 & 19,VAdRS condensers) to receive desorbed liquid condensed by the condenser. The outer chamber covers the inner chamber. The outer chamber consisted of two pipes. The two pipes were placed on the lower and upper sides of chamber (shown in figure 10).In the outer chamber, one end is connected to condenser outlet for bed-1 and similar for bed-2,outer chamber end connected to the ETC solar thermal collector outlet. The hot water is circulated through the generators to adsorption bed (1 & 2 both)60^{o}C temperature of hot water from condenser caused the activated carbon to desorb the refrigerant in bed-1.Then refrigerant vapor was condensed with the help of a condenser and it was stored in the evaporator. The evaporator was placed inside the chilling chamber. A similar event happened in bed-2,during day time temperature of collector increased above 80^{o}C and hot water cause the desorb the R134a from activated carbon.

For bed-1,valves v1 and v2 open for desorption, condensation and evaporation, and other hand valve v3 kept in close position. During adsorption of refrigerant in adsorbent, v3 is open, and v1 & v2 are closed. For bed-2, during daytime valves v4 and v5 open for same desorption, condensation and evaporation process and other hand valve v6 kept in close position. After sunset, collector temperature has been decreased to ambient temperature, the valves v4 & v5 were closed and the valve v6 kept open to allow refrigerant for adsorption process. Valves v7 & v8 regulate hot and cold water respectively from solar collector to adsorption tank and its vice-versa is valid. Desorption process occurs here during day time and adsorption process took place during night. The provision of other valves v9 & v10 are specific, here, solar integrated VAdRS does not work during night, so hot water from condenser can help for desorption of refrigerant from adsorbent for cooling process. Valve v9 allows hot water from condenser and v10 to pass the cold water from adsorption tank or bed-2 and water mixed with feedwater pump line and pumped to the Boiler. This circuit is important part of proposed model for effective cooling during night time.

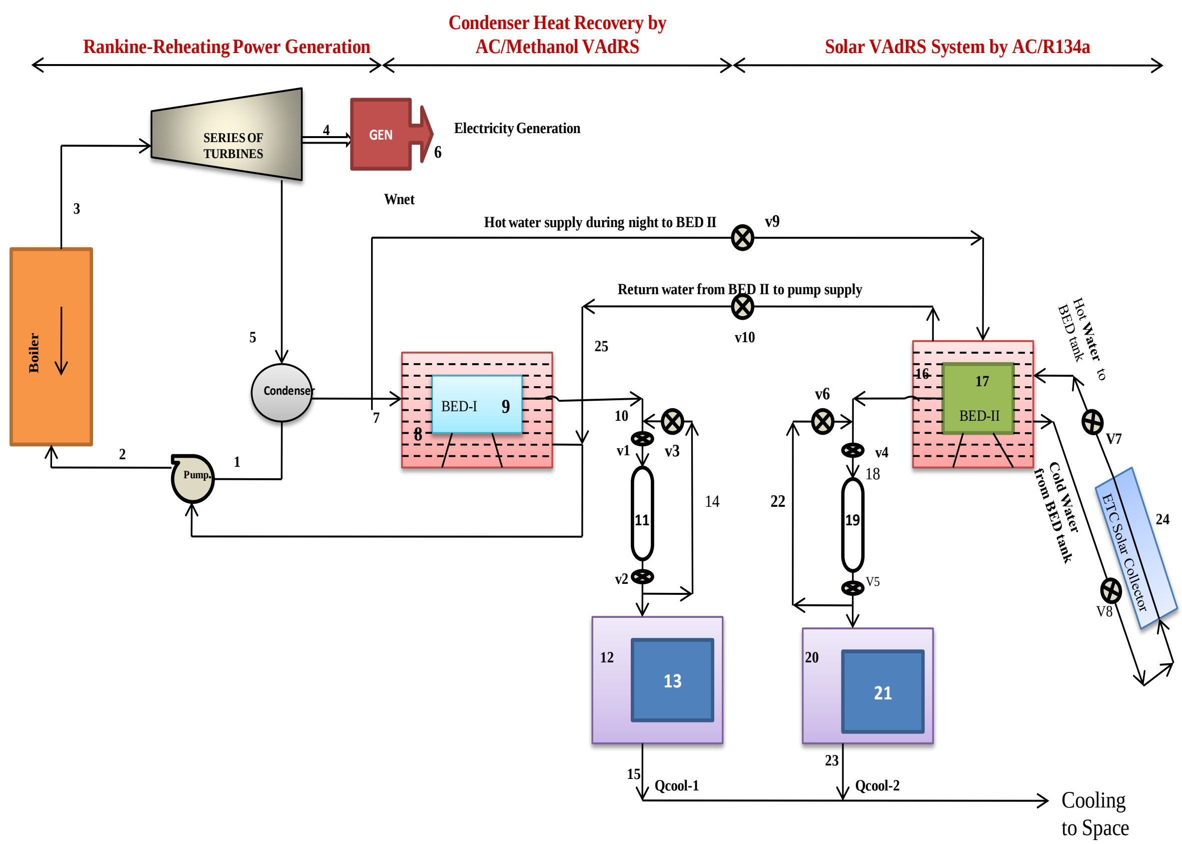


Figure 10: Schematic of combined power and VADRS system

Table 4: Specifications of thermal system

UNIT NO	SPECIFICATION	UNIT NO	SPECIFICATION
1	Condensed steam feed to pump	21	Chiller get to be cooled during evaporation
2	Feed water supply to Boiler for steam formation	22	Evaporated refrigerant adsorbed in adsorbent bed
3	Steam from Boiler supply to series of turbine	23	Chilled air supply to space where cooling is required through cooling line.
4	Turbine shaft work for electricity generation	24	ETC solar thermal water heater for hot water supply to BED-2.
5	Exhaust steam from turbine supply to condenser unit for condensation process	25	Return cold water from BED-2 and mixed with water line for pumping to Boiler.
6	Generator work	v1	Valve 1 open and allow to vaporized refrigerant into condenser for condensation
7	Hot water proceeds to BED-1 and only during night to BED-2 for adsorption cooling process.	v2	Valve 2 allow Condensed refrigerant into evaporator and chiller unit.
8	Outer chamber of BED-1 for hot water collection, (need of	v3	Valve 3 open for adsorption of vapor refrigerant into adsorbent bed after

	hot water for adsorption of heat from adsorbate).		evaporation.(v1 & v2 remain closed when adsorption is happening)
9	Inner chamber of bed -1 , where Activated Carbon-Methanol pair is filled , it utilizes condenser heat.	v4	Valve 4 open and allow to vaporized refrigerant into condenser for condensation
10	Vapor refrigerant (Methanol) supply to condenser of VAdRS after desorption process.	v5	Valve 5 allow Condensed refrigerant into evaporator and chiller unit.
11	Refrigerant liquefy by using condenser	v6	Valve 6 open for adsorption of vapor refrigerant into adsorbent bed after evaporation.(v1 & v2 remain closed when adsorption is happening)
12	Liquid refrigerant get evaporation by evaporator	v7	Valve 7 open for hot water supply from ETC to BED-2 for desorption of refrigerant.(V9 & V10 are closed)
13	Chiller get to be cooled during evaporation	v8	Valve 8 open for cold water return in to ETC after desorption. (V9 & V10 are closed)
14	Evaporated refrigerant adsorbed in adsorbent bed	v9	V9 works during night, when ETC does not work. It opens for hot water supply from condenser exhaust to BED-2.
15	Chilled air supply to space where cooling is required through cooling line.	v10	V 10 allows return cold water from BED-2 after adsorption and water

			mixed with water line for pumping into Boiler.
16	Outer chamber of BED-2 for hot water collection, (need of hot water for adsorption of heat from adsorbate).	Turbine	Series of high-pressure turbine and low-pressure turbine.
17	Inner chamber of bed -2, where Activated carbon-R134a pair is filled, it utilizes condenser heat.	GEN	Generator for electricity generation.
18	Vapor refrigerant (R134a) supply to condenser of VAdRS after desorption process.	BOILER	Steam formation thermal utilities
19	Refrigerant liquefy by using condenser	CONDE NSER	Condensation of exhaust steam and refrigerant vapor condensation process

Model-3 Stack flow heat recovery of combined GT-ST plant using LiBr-H_2O vapor absorption cooling system

Stack flow heat recovery for cooling generation

Exhaust temperature of flue gases from plant is estimated 240-250oC. This temperature has good potential for cooling generation by using vapour absorption cooling machine (VAM) (Figure 11 and table 5) as heat recovery of stack flow. The stack flow temperature must be maintained above 150oC, because the proposed VAM operates with the LiBr-Water system. The input temperature for generator of VAM is maintain between 120-150 0C with 15-20 kg/s of hot water flow from heat exchanger of stack flow (the outlet temp of stack flow from Hex is maintained between 180-200oC).The describe condition of temp and flow rate of water is selected as per cooling requirement, the LiBr-Water system of VAM is designed for 5TR of cooling capacity.

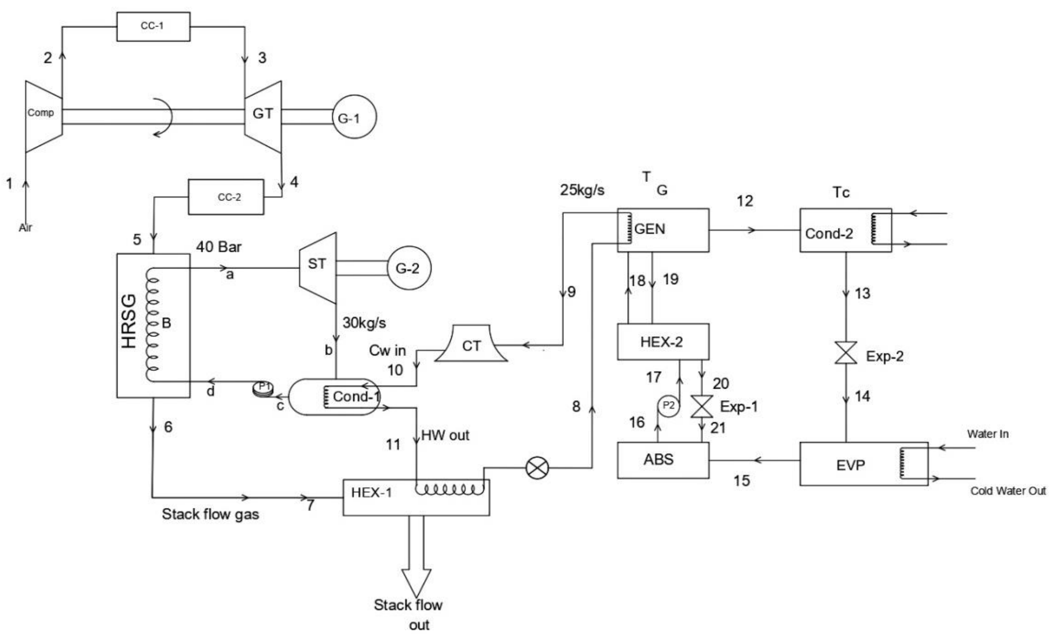


Figure 11: Schematic of combined GT-ST and VARS system

Table 5: Parameters of plant component

PLANT COMPONENTS/PARAMETERS	**UNIT & VALUE**	**PLANT COMPONENTS/PARAMETERS**	**UNIT and VALUE**
Compressor inlet condition (P1,T1)	1bar,25^{o}C	Specific heat ratio of air (ai$_{r}$)	1.4
Compressor pressure ratio (P2/P1)	8	Steam condition at inlet of Steam turbine	40bar,425 ^{0}C

Gas turbine inlet gas temperature (T3)	900^0C	Condenser pressure	0.04 bar
Pressure drop in combustion chamber	3%	Feed water temp to HRSG	170.4^0C
Compressor efficiency (η_c)	88%	ST efficiency (η_{ST})	82%
GT efficiency (η_{GT})	88%	Pressure drop of gas in the HRSG	5kPa
Calorific value of liquid Octane as fuel (CV_f)	44.43MJ/Kg	Steam flow rate= w_s	105 TPH
Specific heat of air (Cp_air)	1.006 KJ/KgK	GT outlet pressure (P4)	1.05 bar
Specific heat of gas (Cp_gas)	1.148 KJ/KgK	Condenser Inlet steam or ST outlet flow	105TPH or 30Kg/s
Specific heat ratio of gas (γ_gas)	1.33		

Model-4 Regeneration-reheating Rankine power generation with eco-friendly refrigerant based vapor jet refrigeration for heat recovery

Regenerative Rankine cycle (figure 12).

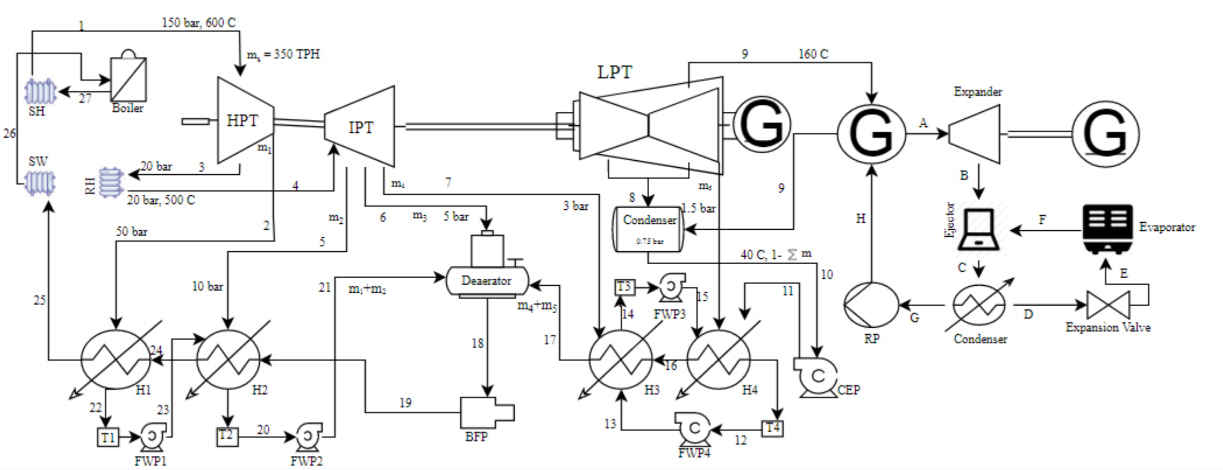


Figure 12: Combined regeneration -reheating Rankine power cycle with VJR system

CHAPTER 4

Chapter-4 Thermodynamic analysis

Introduction

The chapter's present title covers the detail study of all proposed thermodynamic systems and its thermal analysis. The four thermal systems have planned for heat recovery of power plant and capacitive power consumption industry with effect of heating-cooling and power generation. The thermodynamic analysis of systems deals 1st law and 2nd law of thermodynamics (energy conversion & entropy generation), the mathematical optimization techniques has implemented in terms of Taguchi, multi linear regression method of parametric optimization, comparative study. The all thermodynamic system has projected systematically in this chapter.

Research analysis and methodology

Thermodynamic analysis- energy- exergy analysis

4.1 Model-1 Combined cooling heating and power generation by ORC heat recovery with solar thermal Field

Figure 13 and Table 6 shows the schematic and description of combined cooling heating and power generation by ORC heat recovery with solar thermal field.

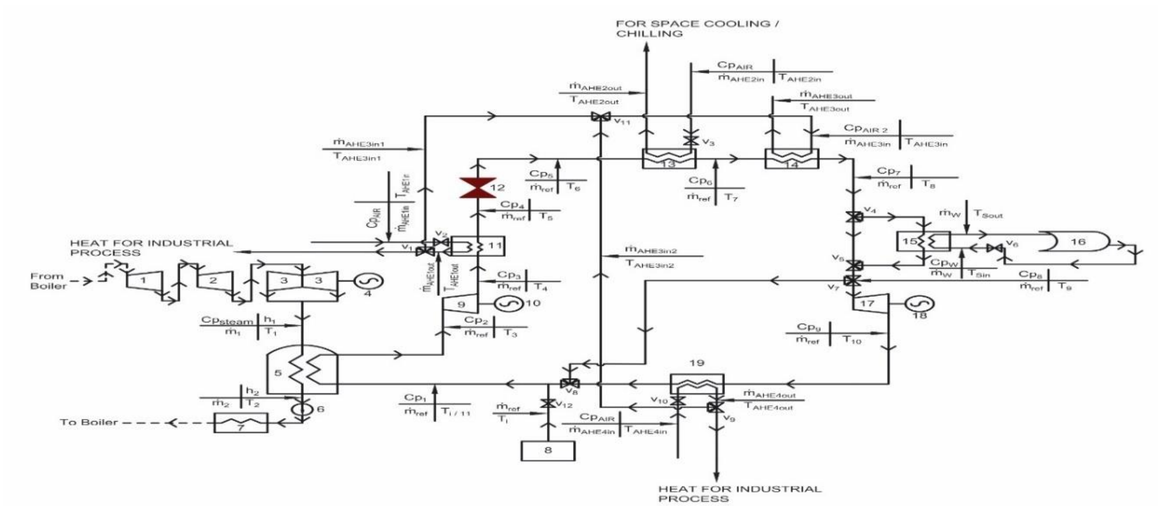


Figure 13: Schematic of combined cooling, heating and power generation plant using condenser heat recovery

Table 6: Description of systems used in the schematic (figure: 13)

System number	**System description**
1	High pressure turbine
2	Intermediate pressure turbine
3	Low pressure turbine
4	Power generator
5	Condenser
6	Circulation extraction pump
7	Feedwater Heater
8	Refrigerant tank
9	Expander 1
10	Power generator
11	Heat exchanger 1
12	Throttle
13	Heat exchanger 2
14	Heat exchanger 3
15	Solar heat exchanger
16	Solar heater
17	Expander 2
18	Power generator
19	Heat exchanger 4
V1	Three-way flow control valve
V2	Two-way flow control valve

V3	Two-way flow control valve
V4	Three-way flow control valve
V5	Three-way flow control valve
V6	Two-way flow control valve
V7	Three-way flow control valve
V8	Three-way flow control valve
V9	Three-way flow control valve
V10	Two-way flow control valve
V11	Three-way flow control valve
V12	Two-way flow control valve

4.2 Model-2 Thermodynamic (energy-exergy) analysis of combined coal based thermal power plant and solar integrated double bed vapor adsorption system for heat recovery and space cooling

Figures 14 and 7 show the schematic and description of Thermodynamic (energy-exergy) analysis of combined coal-based thermal power plant and solar integrated double bed vapor adsorption system for heat recovery and space cooling system.

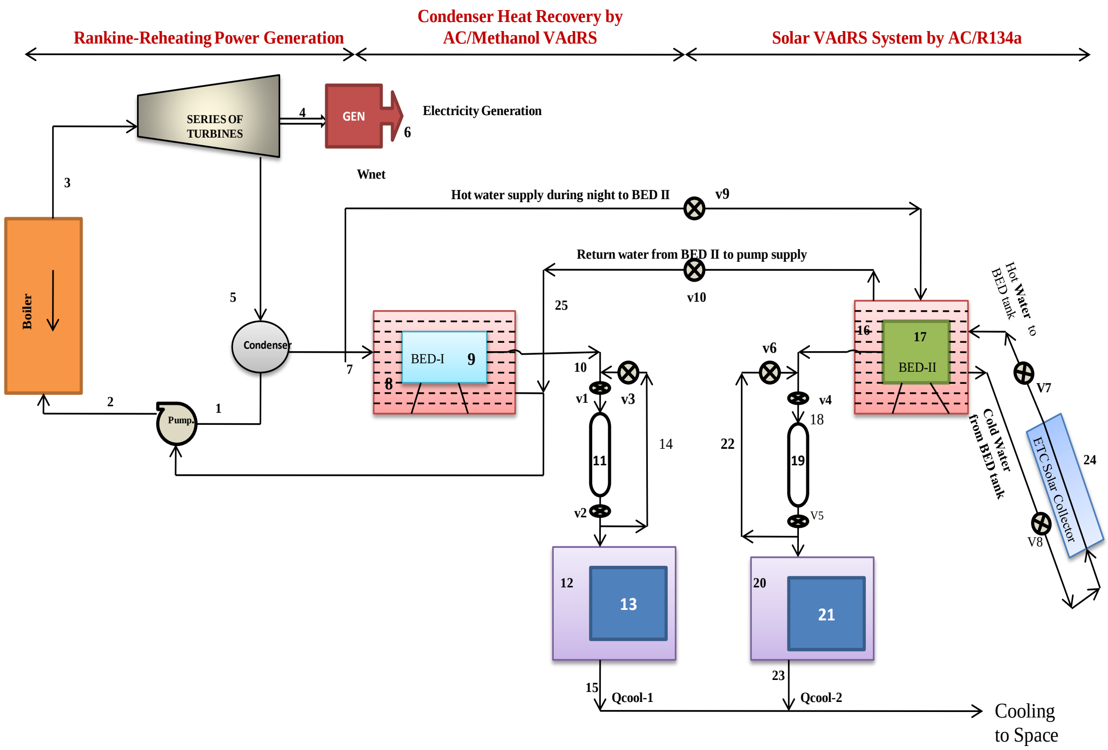


Figure 14: Schematic of combined power and VADRS system

Table 7: Combined power system parts.

UNIT NO	SPECIFICATION	UNIT NO	SPECIFICATION
1	Condensed steam feed to pump	21	Chiller get to be cooled during evaporation
2	Feed water supply to Boiler for steam formation	22	Evaporated refrigerant adsorbed in adsorbent bed
3	Steam from Boiler supply to series of turbine	23	Chilled air supply to space where cooling is required through cooling line.
4	Turbine shaft work for electricity generation	24	ETC solar thermal water heater for hot water supply to BED-2.
5	Exhaust steam from turbine supply to condenser unit for condensation process	25	Return cold water from BED-2 and mixed with water line for pumping to Boiler.
6	Generator work	v1	Valve 1 open and allow to vaporized refrigerant into condenser for condensation

7	Hot water proceeds to BED-1 and only during night to BED-2 for adsorption cooling process.	v2	Valve 2 allow Condensed refrigerant into evaporator and chiller unit.
8	Outer chamber of BED-1 for hot water collection, (need of hot water for adsorption of heat from adsorbate).	v3	Valve 3 open for adsorption of vapor refrigerant into adsorbent bed after evaporation.(v1 & v2 remain closed when adsorption is happening)
9	Inner chamber of bed -1, where Activated Carbon-Methanol pair is filled, it utilizes condenser heat.	v4	Valve 4 open and allow to vaporized refrigerant into condenser for condensation
10	Vapor refrigerant (Methanol) supply to condenser of VAdRS after desorption process.	v5	Valve 5 allow Condensed refrigerant into evaporator and chiller unit.
11	Refrigerant liquefy by using condenser	v6	Valve 6 open for adsorption of vapor refrigerant into adsorbent bed after evaporation.(v1 & v2 remain closed when adsorption is happening)
12	Liquid refrigerant get evaporation by evaporator	v7	Valve 7 open for hot water supply from ETC to BED-2 for desorption of refrigerant.(V9 & V10 are closed)
13	Chiller get to be cooled during evaporation	v8	Valve 8 open for cold water return in to ETC after desorption. (V9 & V10 are closed)
14	Evaporated refrigerant adsorbed in adsorbent bed	v9	V9 works during night, when ETC does not work. It opens for hot water supply from condenser exhaust to BED-2.
15	Chilled air supply to space where cooling is required through cooling line.	v10	V10 allows return cold water from BED-2 after adsorption and water mixed with water line for pumping into Boiler.
16	Outer chamber of BED-2 for hot water collection, (need of	Turbine	Series of high-pressure turbine and low-pressure turbine.

	hot water for adsorption of heat from adsorbate).		
17	Inner chamber of bed -2, where Activated carbon-R134a pair is filled, it utilizes condenser heat.	GEN	Generator for electricity generation.
18	Vapor refrigerant (R134a) supply to condenser of VAdRS after desorption process.	Boiler	Steam formation thermal utilities
19	Refrigerant liquefy by using condenser	Condenser	Condensation of exhaust steam and refrigerant vapor condensation process

4.2.1 Energy-Exergy analysis of combined heating-power and adsorption refrigeration system

Following assumption have been taken in the thermodynamic analysis of proposed combined heating-power and cooling system

A. A control volume oof system is considered, steady state process and constant mass flow rate for the analysis

B. Isentropic performance of pump, turbine and compressor have been taken

C. The efficiency of both turbines and pump assumed isentropic for analysis

D. The refrigerant is adsorbed uniformly in the adsorber and is liquid in the adsorbent

E. The heat loss due to conduction from the adsorber to the condenser and to the evaporator through the metal is neglected.

F. The evaporating liquid temperature should be equal to the evaporator.

All equations of analysis are fundamental approach of 1st law and 2nd law of thermodynamics. And properties (enthalpy, entropy, specific heat, specific volume, etc.) of steam formation referred from steam property table Data [17]as per steam condition (superheated and saturated) and working refrigerants fluid at the different pressure-temperature level. [17]and[18] given energy-exergy equations for power-plant analysis as given below.

The thermal efficiency of plant is expressed by following equation

$\eta_{PLANT} = W_{NET}/Q\_{BOILER}$…………………………………………………………………..(1)

Where Network done by plant is expressed as

$W_{NET}=W_{TURBINE\_TOTAL}-W_{PUMP}$………………………………………………………….. (2)

The Total work done by Turbine is an addition of work one by each turbine as given below

$W_{TURBINETOTAL}=W_{TURBINE\_1}+W_{TURBINE\_2}+W_{TURBINE\_3}$………………………………… (3)

$W_{TURBINE\_1}$=m_dot_turbine1_steamflow(h_tempinlet_turbine1-h_tempexit_turbine1). (4)

$W_{TURBINE\_2}$=m_dot_turbine2_steamflow(h_tempinlet_turbine2-h_tempexit_turbine2... (5)

$W_{TURBINE\_3}$=m_dot_turbine3_steamflow (h_temp inlet_turbine3- h_{2s})………………, (6)

And,

Work done by pump=W_{PUMP}=v×dP= V ×($P_{BOILERLINE}$-$P_{CONDENSERLINE}$)= m_dot_pump (h4s-hf3)………………………………………………………………………………. (7)

From enthalpy and entropy balance equation the isentropic enthalpy at the inlet of condenser (h_{2s}) is expressed as

$h_{2s}=h_{f2}+x_2h_{fg2}$……………………………………………………………………………….. (8)

x_2 is steam dryness fraction at condenser

$s1=s_{2s}=s_{f2}+x_2\ s_{fg2}$……………………………………………………………………… (9)

Heat generated by boiler

Q_{BOILER}=m_dot_Boiler_steam_flow(h_Boilertemp-h) ……………………………… (10)

The energy balance in the condenser is expressed by the following equations

$W_{TURBINETOTAL}-W_{PUMP}=Q\_{BOILER} - Q\_{CONDENSER}$……………………………..... (11)

$\eta_{Boiler}= Q\_{BOILER}/ Q\_{FUEL}$………………………………………………………………(12)

WhereQ_{FUEL}=(m_air+m_fuel)×fuelcalorificvalue ……………………………….....(13)

Thermodynamic modeling of double bed vapors adsorption refrigeration system

The heat of adsorption and desorption balance gives the total heat for the adsorption system, this heat define by the isosteric heating, the following expression as follows [8].

The total energy input to the system is given by

$$Q_{heat\_total}=Q_{isosteric}+ Q_{desorption} \quad (14)$$

The total heat generated from vapour adsorber bed is given below [16]

$$Q_{heat\_total} = \sum m_{ad} [(Cp_{ad} + Cp_r*X_{max}) (T_g - T_{ad}) + (Cp_{ad} + \Delta X/2) (T_{des} - T_g) + \Delta X*H_D]_{BED\text{-}1\ \&\ BED\text{-}2} \quad (15)$$

After condensation of desorbed refrigerant, the liquid refrigerant in the evaporator will absorb the heat of vaporization from liquid to be cooled, this cause the refrigeration effect and the refrigeration amount can be estimate as-

$$Q_{Ref} = \sum [(m_{ad} *\Delta X * L_E)]_{BED\text{-}1\ \&\ BED\text{-}2} \quad (16)$$

The amount of energy used in cooling the evaporated adsorbate when passing through the condenser is given by-

$$Q_{Cool}= \sum [(m_{ad} * \Delta X * Cp_r (T_C-T_E)]_{BED\text{-}1\ \&\ BED\text{-}2} \quad (17)$$

The useful cooling of vapor adsorption system can be estimated by

$$Q_{cool\ useful} = \sum m_{ad} [(L_E * \Delta X - Cp_r (T_C-T_E)]_{BED\text{-}1\&2} \quad (18)$$

Performance parameters involved in the double bed vapour adsorption refrigeration system

The compactness of system is important parameter, Useful cooling per unit mass (specific cooling power)of adsorption system decide the compactness or size of system. Desorption and Adsorption time (second) play a key role in the cooling effect in the VAdRS and its SCP sizing.

$$\mathbf{SCP_{VADRS}} = Q_{cool\ useful} / m_{ad} \quad (19)$$

Or coefficient of performance of VAdRs (is a ratio of useful cooling and total heat input for adsorption and desorption account the refrigeration capacity) is given by following expression

$$\mathbf{COP_{VADRS}} = Q_{cool\ useful} / Q_{heat\_total} \quad (20)$$

Similarly, the solar thermal system COP is a ratio of refrigeration effect of solar vapor adsorption refrigeration system to the solar heat generation

$$COP_{SOLAR} = Q_{cool\ useful} / Q_G \quad (21)$$

Where $Q_G = I * Ac$

Exergy analysis of combined thermal and vapor adsorption plant

Exergy flow rate of inlet steam

$$€_{f\_1n} = m\_dot\_steam \times cp\_Boiler\ steam \times To\ [\frac{Tboiler}{To} - 1 - ln\frac{Tboiler}{TO}] \quad (22)$$

Exergy flow rate of exhaust steam-

$$€_{f\_out} = m\_dot\_steam \times cp\_Boiler\ steam \times To\ [\frac{Tboiler}{To} - 1 - ln\frac{Tboiler}{TO}] \quad (23)$$

Rate of exergy increase of steam = Exergy utilization rate= $€_{f\_Usefull}$

$$€_{f\_Usefull} = m\_dot\_steam\ [h1-h4s-To\ (s1-s4s)] \quad (24)$$

Exergy Destruction in Boiler=$€_{Des\_Boiler}$ = Rate of exergy increase of steam - Rate of exergy decrease

Exergy flow rate of wet steam to condenser

$$€_{wetsteam\ to\ condenser} = m\_{wet\ steam\ to\ condenser}\ [h2s-hf3-To\ (s_2-sf_3)] \quad (25)$$

Irreversibility estimation of each component-

$$I_{RR} = To\Sigma\Delta S = To\ (\Delta S_{Boiler} + \Delta S_{turbine} + \Delta S_{condenser} + \Delta S_{pump} + \Delta S_{Bed-1} + \Delta S_{Bed-2} + \Delta S_{solar}) \quad (26)$$

$$\text{And } \Delta S = cpln(\frac{Tout}{Tin}) \quad (27)$$

(cp will vary with temperature and pressure condition of working fluid, putting all value of cp and Tout&Tin of components)

$$\text{Now} \quad EDRratio = \frac{To\Sigma\Delta S}{Qboiler} \quad (28)$$

$$\text{Exergetic efficiency} = \eta_{EX} = 1 - EDR_{ratio} \quad (29)$$

$$\text{Exergy Destruction Rate} = \frac{To\Sigma\Delta S}{Wnet} \quad (30)$$

4.3 Model-3 Stack flow heat recovery of GT-ST plant using VARS

4.3.1 Thermodynamic analysis

Stack flow recovery through VARS plant and plant components are given in figure 15 and table8, respectively.

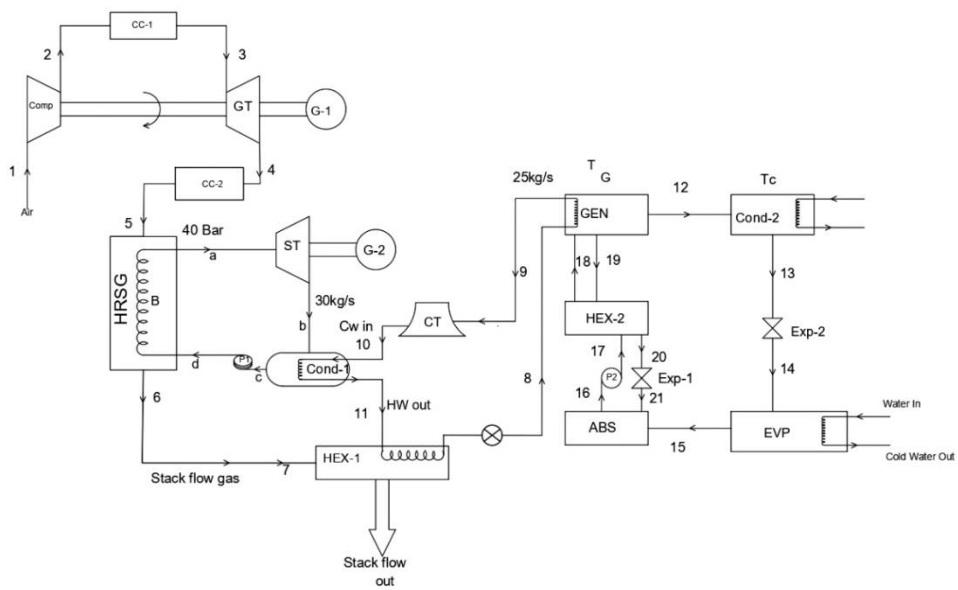


Figure 15: Stack flow heat recovery through VARS plant

Table 8: Plant Component values

Plant components/ Parameters	Unit &value	Plant components/ Parameters	Unit and value
Compressor inlet condition (P1,T1)	1bar,25^0C	Specific heat ratio of air (γ_{air})	1.4

Compressor pressure ratio (P2/P1)	8	Steam condition at inlet of steam turbine	40bar,425 ^{0}C
Gas turbine inlet gas temperature (T3)	900 ^{0}C	Condenser pressure	0.04 bar
Pressure drop in combustion chamber	3%	Feed water temp to HRSG	170.4 ^{0}C
Compressor efficiency (η_c)	88%	ST efficiency (η_{ST})	82%
GT efficiency (η_{GT})	88%	Pressure drop of gas in the HRSG	5kPa
Calorific value of liquid Octane as fuel (CV_f)	44.43MJ/Kg	Steam flow rate= w_s	105 TPH
Specific heat of air (Cp_{air})	1.006 KJ/KgK	GT outlet pressure (P4)	1.05 bar
Specific heat of gas (Cp_{gas})	1.148 KJ/KgK	Condenser inlet steam or ST outlet flow	105TPH or 30Kg/s
Specific heat ratio of gas (γ_{gas})	1.33		

Assume $\psi = (\Delta G_0/\Delta H_0)$ =1.0401 + 0.1728 (h/c)

Where h/c =mass ration of hydrogen to carbon in fuel

GT Plant

Compressor

P1 = 1bar,P2=8 bar,T1=25 ^{0}C =298K and ηc= 0.88

$$\frac{T2}{T1} = (\frac{P2}{P1})^{\frac{(\gamma-1)}{(\gamma * \eta c)}} \tag{1}$$

$T2 = 313\ ^0C = 586K$ (temp after compression)

Combustor

Pressure loss = 3% × (P2/P1) (2)

= 0.03 × 8 =0.24 bar

Pressure after combustor P3 = 8-0.24 = 7.76 Bar

Assume flow rate of combustion gas is 1Kg/s and that of fuel f Kg/s

The flow rate of air = (1-f) Kg/s (3)

Therefore $f \times CV_f = m_f \times Cp\_{gas}(T3-T4) - (1-f)\ Cp\_air\ (T2-T0)$ (4)

f×44430 = 1 × 1.148 (900-25) – (1-f) × 1.006 (313-25)

f = 0.0162 Kg/s

Air fuel ratio = $\frac{A}{F} = \frac{(1-f)}{f}$ (5)

A/F = 60.73

Octane combustion – C8H18 + 12.5 O2 = 8CO2 + 9H20 (6)

For stoichiometric combustion A/F ratio = $\frac{(12.5 \times 32)}{(0.232 \times 114)}$= 15.12

(C8H18 = 12*8 + 1*18=114)

Excess air = (60.73-15.12/15.12)= 3.02 or 302%

Gas turbine

$P4 = 1.05$ bar, $T3 = 900^0C = 1173K$

$$\frac{T3}{T4} = \left(\frac{P3}{P4}\right)^{\frac{(\gamma-1)(\eta GT)}{(\gamma)}} \quad (7)$$

$T4 = 755K = 482\ ^0C$ (GT exhaust temp)

HRSG

Let the pinch point temp differences (T5-Tf) be 30 ^0C

(Pinch point at exit of GT and inlet of ST)

$Tf = (Tsat)_{40\ bar} = 250.4\ ^0C$

$T5 = 250.4 + 30 = 280\ ^0C$

From steam table –at Ta =ST inlet temp=425 ^0C = h_a=3272 KJ/Kg

h_f=1087 KJ/Kg (at Tf)

From mass and energy balance between GT and ST

$$m\_gas \times Cp\_gas\ (T4-T5) = m\_steam\ (h_a-h_f) \quad (8)$$

If 1 Kg/s of gas flow in GT, then steam flow rate as calculated

$1\times 1.14\ (482-280) = m\_steam\ (3272-1087.3)$

$m\_steam = 0.106$ kg/s

But steam flow rate is given 29.2 kg/s, then gas flow rate is

$m\_gas \times 1.14\ (482-280) = 29.2\ (3272-1087.3)$

$m\_gas = w_g = 275.4$ Kg/s

$$\text{And air flow rate entering the compressor} = m\_air = (1-f)\ m\_gas \quad (9)$$

$m\_air = w_a = 271.3$ kg/s

$$\text{Fuel mass flow rate } m\_fuel = w_f = f\times m\_gas \quad (9a)$$

$w_f = 0.0162 \times 257.4 = 4.46$ Kg/s

The total heat transfer in the HRSG, at Feed water temp=Te=170.4 ^0C

h_e=721.1 KJ/Kg

$1.14\ (482-T6) = 0.106\ (3272-721.1)$

Stack flow temp= T6 or T12=247 ^0C

Power output of GT-ST plant-

Total power output $W\_total = W\_{ST} + W\_{GT}$

$$W\_{ST} = w_s\ (ha - h_{bs}) \times \eta\_st \quad (10)$$

From enthalpy and entropy equation between steam turbine expansion process

$h = h_f + xh_{fg}$ (11)

$s = s_f + x\ s_{fg}$ (12)

At 40 bar-425 ^{0}C (at inlet of ST) and 0.04 bar (at condenser line)

Enthalpy, ha = 3272 KJ/Kg and Entropy s_a = 6.853 Kj/Kg-K

So, $s_{bs} = s_{f\_b} + x\_bs \times s_{fg\_b}$ (13)

(All entropy values from steam table at 0.04 bar pressure)

s_a = 6.853 Kj/Kg-K = s_{bs} = 0.4226 +x_bs × 8.052

Steam quality after expansion = x_bs = 0.7986 or 80%

And enthalpy at condenser line = $h_{bs} = h_{f\_b} + x\_bs \times h_{fg\_b}$ (14)

h_{bs}= 121.46+0.798× 2431.9

h_{bs}= 2064.3 kJ/kg

From equ (9) $W_{\_ST}$ = 29.2 (3272– 2064.3)× 0.82

$W_{\_ST}$ = 29MW

$W_{\_GT}$= m_gas × Cp_gas (T3-T4) -m_air × Cp_air (T2-T1) (15)

$W_{\_GT}$= 275.4×1.14 (900+273-482+273)-271×1.006(313+273-25+273)

$W_{\_GT}$= 53MW

Total plant output = W_total = 29+53=82MW

When two plants are combined, there is always some heat loss. If heat rejected by GT plant as topping cycle is absorbed by ST plant as bottoming cycle, the Lost heat as coefficient in the exhaust stack

$X_L = \left(\frac{\mathbf{wg \times Cpg(T6-T1)}}{\boldsymbol{wf \times CVf}}\right)$ (16)

X_L=(275.4×1.14×(247+273-25+273)/4.46×44430)

X_L=0.354

The overall efficiency of plant, when considering heat lost

$\eta_{overall}= \eta_{ST}+\eta_{GT} - \eta_{ST}\times\eta_{GT}-\eta_{ST}\times X_L$ (17)

η_{ST}= (ha-hb)/(ha-he) (18)

η_{ST}= (3272-2064)/(3272-721)=0.38 0r 38%

$\eta_{STplant}$=38%

$\eta_{GTplant}$ = $(W_{GT})/(w_f\times CV_f)$ (19)

η_{GT} = (53MW)/(4.46×44430)=0.27 Or 27%

From Equ No 17 $\eta_{overall}$ = 0.38+0.27–0.38×0.27-0.38×0.354

$\eta_{overall\ GT\text{-}ST\ plant}$ = 41.7%

Exergy analysis

Assume exergy flux $\psi = (\Delta G_0/\Delta H_0) = 1.0401 + 0.1728\ (h/c)$ (20)

Where, (h/c) – mass ratio of hydrogen to carbon in Octane (C8H18) fuel

$\psi = (\Delta G_0/\Delta H_0) = 1.0401 + 0.1728\ (18\times1/8\times12)$

$\psi = (\Delta G_0/\Delta H_0) = 1.072$

$\Delta H_0 = w_f\times(CV)_0$=4.46×44430=198424 kW (21)

From Equation no 20

$\Delta G_0=\psi\times\Delta H_0$=1.072×198.424 kW = 212810 kW =Exergy input

$T_0\Delta S_0 = \Delta G_0-\Delta H_0$ (22)

$T_0\Delta S_0$= 212810-198424=1438 6kW

Exergy destruction of components due to irreversibility

Compressor

Rate of energy dissipation in compressor (I_{comp}) =w_aT_0 (s2-s1) (23)

Now, s2-s1=$Cp_a \ln \frac{T2}{T1} - R_a \ln \frac{P2}{P1}$ (24)

But $R_a = C_p\ (\frac{\gamma-1}{\gamma})$ (25)

$s2\text{-}s1 = 1.006\ ln\frac{(313+273)}{298} - \frac{(0.4\times1.006)}{1.4}\ ln8$

s2-s1=0.0818kJ/kg-k

$I_{comp}=271 \times 298\times0.0818$

***I*$_{comp}$=6613 kW**

Combustion chamber

Exergy destruction in combustion chamber is estimated by the energy balance between product, reactants, air and used fuel respectively.

$Icc = T_0 [(S_P)_3-(S_R)_2$ (26)

$Icc = T_0 [\{(S_P)_3-(S_P)_O\}+\{(S_A)_2-(S_A)_O\}+\Delta S_0$ (26a)

$Icc = T_0\left[\left\{wgCpg\ ln\frac{T3}{To} - WgRg\ ln\frac{P3}{Po}\right\} - \left\{wgCpg\ ln\ \frac{T2}{T0} - waRa\ ln\ \frac{P2}{Po}\right\} + \Delta S\right]$ (26b)

$Icc = T_0\left[wg\left\{Cpg\ ln\frac{T3}{To} - Rg\ ln\frac{P3}{Po}\right\} - \left\{wgCpg\ ln\ \frac{T2}{T0} - waRa\ ln\ \frac{P2}{Po}\right\} + \Delta S\right]$ (26c)

$Icc=298\left[275.8\left\{1.148ln\frac{1173}{298} - \frac{1.148}{4}\ln 7.76\right\} - \left\{271.4\times1.006\ ln\ \frac{586}{298} - 271.3\times \frac{1.006}{3.5}\ ln\ 8\right\}] + To\Delta S\right]$ (27)

***I*cc = 80947-6672+14386=88661 kW**

Gas turbine

Rate of energy lost or work lost in GT = $I_{GT}=w_gT_0\ (s_4-s_3)$ (28)

Where $s_4-s_3 = \mathrm{Cpg}\ln\frac{T4}{T3} - \mathrm{Rg}\ln\frac{p4}{p3}$ (29)

$I_{GT}=w_gT_0\ [\mathrm{Cpg}\ln\frac{T4}{T3} - \mathrm{Rg}\ln\frac{p4}{p3}]$ (30)

$I_{GT} = 275.8 \times 298\ [-1.148\ ln\ (1.533) - \frac{1.148}{4}\ ln\frac{1.05}{7.76}]$

***I*$_{GT}$ =5646 kW**

HRSG

Rate of energy lost in heat recovery steam generator-I_{HRSG}

I_{HRSG} = $T_0[ws(Sa - Se) + wg(s6 - s4)]$ (31)

I_{HRSG} = $T_0[ws(sa - se) + Cpg\ ln\frac{T6}{T4} - Rg\ ln\ \frac{P6}{P4}]$ (31a)

I_{HRSG} = $298[29.235(6.853 - 2.046) + 1.148 \ln\frac{755}{520} - \frac{1.148}{4} \ln\frac{105}{100}]$

***I*HRSG = 7853 kW**

Steam turbine

Rate of energy or work lost in the steam turbine= I_{ST}

$I_{ST} = T_0\ w_s\ (s_b\text{-}s_a)$ (32)

[Using equation, no (11) & (12)]

For $s_b = s_b = (s_f + x\ s_{fg})_b$ (33)

But steam quality at state 'b'

$h_b = (h_f + xh_{fg})_b$ (34)

But $h_a\text{-}h_b = \eta_{ST}\ (h_a\text{-}h_{bs})$ (35)

From Steam properties table

s_a = 6.853 kJ/kG-K, h_a = 3272 kJ/kG, h_{bs} = 2064.37kJ/kG

And η_{ST} = 82% = 0.82 (taken data)

Put all values in equation no (35)

x_b = 0.89

From equation (33), s_b = 0.422 + 0.89 ×8.052 =7.586 kJ/kG-K

Now, I_{ST} = 29.235 (7.586-6.853)

$\boldsymbol{I_{ST}}$ **= 6412 kW**

Exergy lost due to exhaust flue gas

$$I_{EXHFLUEGAS} = \int_{T0}^{T6} \left(1 - \frac{T0}{T}\right) dQ \tag{36}$$

$$= w_g \times Cpg\,[(T6 - T0) - T0\,ln\frac{T6}{T0}] \tag{36a}$$

$$=275.8 \times 1.148\,[(247 + 273 - 25 + 273) - 298\,ln\,\frac{520}{298}]$$

$\boldsymbol{I_{EXHFLUEGAS}}$ **= 17760kW**

Exergy balance of GT-ST power plant

Table 9: Exergy of GT-ST plant component

Components	**Exergy losses(kW)**	**Components**	**Exergy losses(kW)**
Compressor	6613	Heat recovery steam generator	7583
Combustion chamber	88661	ST	6412
GT	5646	Stack flow	17760
Exergy input (ΔG_0)	212810 kW	**Total output/Total I_{LOSS}**	82MW or 82000kW/132945

$$\text{Exergetic efficiency}=\eta_{EX}=\frac{Total\ Output}{Exergy\ Input} \tag{37}$$

$\eta_{EX}=\frac{82000}{212810}$ = 0.385 =**38.5%**

[Exergy output+Exergy destruction = 82000+132945 = 214945 kW, means it's greater than exergy input]

Stack flow heat recovery for cooling generation

Exhaust temperature of flue gases from plant is estimated 240-250 ^{0}C. This temperature has good potential for cooling generation by using vapour absorption cooling machine (VAM) as heat recovery of stack flow. The stack flow temperature must maintain 150 ^{0}C because when coal is combusted, more than 95% of the Sulphur is converted to SO2, and a small fraction gets converted to gaseous SO_3. However, when the flue gas temperature drops below315 ^{0}C, SO3 begins to react with the water vapor to yield sulphuric acid vapor. This reaction is completed when the temperature drops to about 150–175 ^{1}C, the temperature at which precipitators normally operate. The proposed VAM is operating with LiBr-Water system. The input temperature for VAM generator is maintained between 120-150 ^{0}C with 15-20 kG/s of hot water flow from heat exchanger of stack flow (the outlet temp of stack flow from Hex is maintained between 180-200 ^{0}C).The describe condition of temp and flow rate of water is selected as per cooling requirement, the LiBr-Water system of VAM is designed for 5TR of cooling capacity.

Heat reject by stack flow = Heat gain by condenser circulated water for VAM generator(38)

The 30-40^{0}C of water (water from cooling tower at 20^{0}C and gained heat from outlet steam from ST) comes from condenser and circulated inside the stack flow HEx.

$$w\_gas \times Cp\_gas\ (T_{stack\_in} - T_{stack\_out}) = w\_cw \times Cp\_water\ (T_{Hot\_water} - 298) \quad (38a)$$

For maintaining 40^{0}C of temp at condenser outlet to stack flow, the cooling tower water flow rate must be maintaining and mass balance equation is applying as

Water flow from cooling tower

$$m_s(c_p\Delta t + L.H)_{cond} = m_{cw}(c_p\Delta t)_{c.t\ side} \quad (38b)$$

$$30\ (2.12 * 100 + 330) = m_w\ (4.18 * 40)$$

(m_s is steam flow rate as 30Kg/s)

m_{cw}= 95 kg/s (For maintain 40^0C of hot water flow at out of condenser to cooling tower. But for stack flow Hex non-return valve is used and maintain generator temp by 110 to 165 ^0C with flow between 25 to 40 Kg/s to generator of VARSplant)

$$275.8 \times 1.148\,(247 + 273 - 200 + 273) = 25 \times 4.184(\mathrm{THot\_water} - 25 + 273)$$

T_{Hot_water} = 165^0C(VAM Generator temp source, Generator temp is taken as 110^0C as per refrigerant property)

5TR L_IB_R-H_20 VAM analysis

Assumptions-

T_G=Generator Temperature

T_C=Condenser Temperature

T_A=Absorber Temperature

T_E=Evaporator Temperature

P_G=Generator Pressure

P_C=Condenser Pressure

P_A=Absorber Pressure

P_E=Evaporator Pressure

ξ = 1 for refrigerant

$\xi ss = \xi r$ = rich solution (flow from Gen to Cond)

$\xi ws = \xi a$ =week solution (flow from Gen to Abs)

(Sp heat for ξa and ξr will be same)

Condenser pressure maintained at water vapour pressure because water vapour flows from Gen to Cond at same pressure.

Operating Condition

P_G=P_C=15.75 kPa at T_G=110 ^{0}C(from refrigerant property table)

P_A=P_E=0.8721 kPa at T_E=5 ^{0}C (from refrigerant property table)

T_A= 35^{0}C and T_C=55 ^{0}C

Condenser

At 55 ^{0}C saturated water vapour pressure, properties from steam table

At 55 ^{0}C, P_C=15.75 kPa, h_6=h_f=230 kJ/kG

H_state =Hg +0.01 ^{0}C +Cp of saturated Vapour (39)

This equation is valid for only enthalpy of saturated vapour

From h-s plot of cooling cycle-h_6=h_7=230 kJ/kG

Evaporator

At T_E=5 ^{0}C, P_E=0.8721 kPa, h_8=h_{g8} = 2510.6 Kj/Kg

Cooling Plant Refrigeration Effect (Q_{REF})= 5TR

(1TR=3.5kW)

Q_{REF} = 5TR = 5×3.5=17.5kW

Q_{REF} = m_ref (h_8-h_7) (40)

17.5 kW = m_ref (2510-230)

Flow rate of refrigerants = m_ref = 0.008 kG/sec 0r 470 gram/sec

(Water vapour distilled during refrigeration)

Generator

At T_G=110 ^{0}C, h5 = 2676 kJ/kG

Heat Rejected by Condenser (Q_{COND}) = m_ref (h_5-h_6) (41)

Put all values, Q_{COND} = 19.5kW

Circuit Energy Balance-

Circulation Ratio **(f) =** $\boldsymbol{f} = \frac{(1-\xi r)}{(\xi r-\xi a)} = \frac{\xi ws}{\xi ss-\xi ws}$ (42)

At Absorber-P_A=P_E=0.8721 kPa

At, T_A=35 ^{0}C, ξ_{LiBr} = 0.45 (from h-ξ plot) and enthalpy h4 = 70kJ/kG

So, Rich solution (water) = ξ_{REF} = ξss = 1-0.45 = 0.55

At Generator-P_G=P_C=15.75 kPa at T_G=110 ^{0}C

ξ_{LiBr} = 0.50 (from h-ξ plot) and enthalpy h_2 = 250 kJ/kG

Poor solution concentration of water (ξ_{REF}) = 1-0.40 = 0.50

State-1saturated solution at condenser (at ξ_{LiBr} = 0.45)

h_1 = 160 kJ/Kg (Temp below T_G, taken T_{1a} = 70 ^{0}C)

State-3 saturated solution at PE and ξ_{LiBr}= 0.5

T3 = 55 0C, h3 = 130 kJ/Kg

State-3a ξ_{LiBr} = 0.45, Temp = 4 ^{0}C

Solution circulation ratio = $\boldsymbol{f} = \frac{(1-\xi r)}{(\xi r-\xi a)} = \frac{(1-0.55)}{(0.55-0.5)}$ = 9 kG of vapour

Regenerative heat exchanger (REG HEx)

Heat available in hot solution for transfer (Q_{REGHEx_hot})

Q_{REGHEx_hot} = (f-1) (h_2-h_1) (43)

Q_{REGHEx_hot} = (9-1) (250-130) = 960Kj/Kg

Heat requires by cold solution for heat in REG Hex (Q_{REGHEx_cold})

Q_{REGHEx_cold} = f (h_1-h_4) (44)

Q_{REGHEx_cold} = 9(160-70) = 810 Kj/kG

Energy Balance in Reg Hex

$f\ (h_{1a}-h_4) = (f-1)(h_2-h_3)$ (45)

$9(h_{1a}-70) = (9-1)\ (250-130)$

$h_{1a}= 176$ kJ/kG

Heat Generation in Generator (Q_{GEN})

Q_{GEN} = m_ref $[h_5-h_2+f\ (h_2-h_{1a})]$ (46)

$Q_{GEN} = 0.008\ [2676-250+9(250-176)]$

$Q_{GEN} = 24.7$ kW

Absorption of Heat (Q_{ABS})

Q_{ABS} = m_ref $[h_8-h_3+f\ (h_3-h_4)]$ (47)

$Q_{ABS} = 0.008[2510-130+9(130-70)]$

Q_{ABS} = 23.36 Kw

$COP_{LiBr\text{-}H2O\text{-}\_VAM} = COP_I = \frac{Qref}{Qgen}$ (48)

$COP_{LiBr\text{-}H2O\text{-}\_VAM} = \frac{17.5}{24.7}\ = 0.708$

Now, $COP_{II}= COP_1\ /\ COP_{CARNOT}$ (49)

$COP_{CARNOT} = \left(\frac{TE}{TC-TE}\right)(\frac{TG-TA}{TG})$ (50)

$\mathbf{COP_{CARNOT} = \left(\frac{278}{50}\right)(\frac{65}{373}) = 0.97}$

From Equation (49)-$COP_{II}= 0.708\ /\ 0.97$

COP_{II} =0.73

Statistical analysis

4.3.2 Regression analysis: Topping plant heat rejection versus Pressure (Bar), Temperature (Degree Celsius), CP_flue gas, M_flue gas, Temp diff

1. **Overall efficiency**

Regression equation

Overalleffi (%)	=	46.6 + 0.423 Pressure(Bar) - 0.01883 Temperature (Degree Celsius) - 0.798 CP_flue gas + 0.000009 m_flue gas + 0.0260 Temp Diff - 0.0140 Topping plant heat rejection - 0.002 Heat loss (KW) + 0.2270 Boiler effi (%) + 0.9367 Topping plant effi (%) - 0.090 Bottoming plant effi (%)

Coefficients

Term	Coef	SE Coef	T-Value	P-Value	VIF
Constant	46.6	27.4	1.70	0.150	
Pressure(Bar)	0.423	0.211	2.01	0.101	193.75
Temperature (Degree Celsius)	-0.01883	0.00832	-2.26	0.073	3015.17
CP_flue gas	-0.798	0.401	-1.99	0.103	38.17
m_flue gas	0.000009	0.000024	0.39	0.711	16.00
Temp diff	0.0260	0.0121	2.14	0.085	2946.34
Topping plant heat rejection	-0.0140	0.0827	-0.17	0.872	1123.79
Heat loss (KW)	-0.002	0.108	-0.02	0.988	3392.53
Boilereffi (%)	0.2270	0.0694	3.27	0.022	3590.60
Topping plant effi (%)	0.9367	0.0511	18.34	0.000	86.49

Bottoming plant effi (%)	-0.090	0.825	-0.11	0.917	5.04

Model summary

S	R-sq	R-sq(adj)	R-sq(pred)
0.0677665	99.99%	99.98%	99.46%

Analysis of variance

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Regression	10	324.181	32.4181	7059.23	0.000
Pressure(Bar)	1	0.018	0.0185	4.02	0.101
Temperature (Degree Celsius)	1	0.024	0.0235	5.12	0.073
CP_flue gas	1	0.018	0.0182	3.96	0.103
m_flue gas	1	0.001	0.0007	0.15	0.711
Temp Diff	1	0.021	0.0211	4.59	0.085
Topping plant heat rejection	1	0.000	0.0001	0.03	0.872
Heat loss (KW)	1	0.000	0.0000	0.00	0.988
Boilereffi (%)	1	0.049	0.0492	10.72	0.022
Topping plant effi (%)	1	1.544	1.5442	336.25	0.000
Bottoming plant effi (%)	1	0.000	0.0001	0.01	0.917
Error	5	0.023	0.0046		
Total	15	324.204			

Fits and diagnostics for unusual observations

Obs	Overall effi (%)	Fit	Resid	Std Resid		
5	77.5073	77.5108	-0.0034	-0.90		X
8	68.5692	68.5711	-0.0018	-0.37		X
15	81.5514	81.6430	-0.0915	-2.15	R	

R Large residual

X Unusual X

2. **Topping plant efficiency**

Regression equation

Topping Plant Heat Rejection	=	206 + 13.4 Pressure(Bar) - .441 Temperature (Degree Celsius) + 9.68 CP_flue gas - 0.00163 m_flue gas + 0.611 Temp diff

Coefficients

Term	Coef	SE Coef	T-Value	P-Value	VIF
Constant	206	150	1.37	0.200	
Pressure(Bar)	13.4	13.7	0.98	0.351	91.79
Temperature (Degree Celsius)	-0.441	0.436	-1.01	0.336	928.08
CP_flue gas	9.68	6.27	1.54	0.154	1.05
m_flue gas	-0.00163	0.00113	-1.44	0.180	4.07
Temp diff	0.611	0.654	0.93	0.372	962.16

Model summary

S	R-sq	R-sq(adj)	R-sq(pred)
6.39803	45.75%	18.63%	0.00%

Analysis of variance

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Regression	5	345.26	69.05	1.69	0.225
Pressure(Bar)	1	39.12	39.12	0.96	0.351
Temperature (Degree Celsius)	1	41.85	41.85	1.02	0.336
CP_flue gas	1	97.59	97.59	2.38	0.154
m_flue gas	1	85.09	85.09	2.08	0.180
Temp diff	1	35.71	35.71	0.87	0.372
Error	10	409.35	40.93		
Total	15	754.60			

Fits and diagnostics for unusual observations

Obs	Topping plant heat rejection	Fit	Resid	Std Resid	
8	40.98	56.35	-15.37	-2.82	R

R Large residual

3. **Overall effi with effect of topping and bottoming effi with Boiler effi and heat loss**

Regression equation

Overalleffi (%)	=	40.1 + 0.00417 Heat Loss (KW) + 0.21460 Boiler effi (%) + 0.8652 Topping plant effi (%) - 0.119 Bottoming plant effi (%)

Coefficients

Term	Coef	SE Coef	T-Value	P-Value	VIF
Constant	40.1	28.3	1.42	0.184	
Heat loss (KW)	0.00417	0.00863	0.48	0.638	3.93
Boilereffi (%)	0.21460	0.00557	38.53	0.000	4.18
Topping plant effi (%)	0.8652	0.0136	63.42	0.000	1.11
Bottoming plant effi (%)	-0.119	0.886	-0.13	0.895	1.05

Model Summary

S	R-sq	R-sq (adj)	R-sq (pred)
0.159477	99.91%	99.88%	99.74%

Analysis of Variance

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Regression	4	323.924	80.981	3184.11	0.000
Heat loss (KW)	1	0.006	0.006	0.23	0.638
Boiler effi (%)	1	37.754	37.754	1484.47	0.000
Topping plant effi (%)	1	102.285	102.285	4021.76	0.000

Bottoming plant effi (%)	1	0.000	0.000	0.02	0.895
Error	11	0.280	0.025		
Total	15	324.204			

Fits and Diagnostics for Unusual Observations

Obs	Overall effi (%)	Fit	Resid	Std Resid	
13	69.5903	69.2733	0.3169	2.74	R

R Large residual

4. **Overall, with effect of topping and bottoming effi and heat loss**

Regression equation

Overall Effi (%)	=	-172 - 0.2827 Heat Loss (KW) + 1.030 Topping plant effi (%) + 6.93 Bottoming plant effi (%)

Coefficients

Term	Coef	SE Coef	T-Value	P-Value	VIF
Constant	-172	310	-0.55	0.589	
Heat loss (KW)	-0.2827	0.0488	-5.80	0.000	1.01
Topping plant effi (%)	1.030	0.145	7.12	0.000	1.00
Bottoming plant effi (%)	6.93	9.68	0.72	0.488	1.01

Model Summary

S	R-sq	R-sq (adj)	R-sq (pred)
1.78031	88.27%	85.34%	78.82%

Analysis of Variance

Source	DF	Adj SS	Adj MS	F-Value	P-Value

Regression	3	286.170	95.390	30.10	0.000
Heat loss (KW)	1	106.559	106.559	33.62	0.000
Topping plant effi (%)	1	160.723	160.723	50.71	0.000
Bottoming plant effi (%)	1	1.625	1.625	0.51	0.488
Error	12	38.034	3.170		
Total	15	324.204			

Fits and diagnostics for unusual observations

Obs	Overall effi (%)	Fit	Resid	Std Resid	
8	68.569	73.428	-4.858	-3.07	R

5. **Overall effi with effect of topping, bottoming and Boiler effi**

Regression Equation

Overall effi (%)	=	38.2 + 0.8668 Topping plant effi (%) - 0.055 Bottoming plant effi (%) + 0.21228 Boiler effi (%)

Coefficients

Term	Coef	SE Coef	T-Value	P-Value	VIF
Constant	38.2	27.1	1.41	0.184	
Topping plant effi (%)	0.8668	0.0128	67.79	0.000	1.05
Bottoming plant effi (%)	-0.055	0.848	-0.07	0.949	1.03
Boiler effi (%)	0.21228	0.00273	77.85	0.000	1.07

Model Summary

S	R-sq	R-sq(adj)	R-sq(pred)
0.154299	99.91%	99.89%	99.77%

Analysis of Variance

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Regression	3	323.918	107.973	4535.11	0.000
Topping plant effi (%)	1	109.417	109.417	4595.78	0.000
Bottoming plant effi (%)	1	0.000	0.000	0.00	0.949
Boilereffi (%)	1	144.308	144.308	6061.27	0.000
Error	12	0.286	0.024		
Total	15	324.204			

Fits and diagnostics for unusual observations

Obs	Overall effi (%)	Fit	Resid	Std Resid	
13	69.5903	69.2744	0.3158	2.82	R

R Large residual

6. **Overall effi with effect of GT-IT, compression ratio and flow rate of flue gas**

Regression equation

Overalleffi (%)	=	31.79 + 1.161 Pressure(Bar) + 0.01690 Temperature (Degree Celsius) + 11.80 CP_flue gas - 0.000301 m_flue gas

Coefficients

Term	Coef	SE Coef	T-Value	P-Value	VIF
Constant	31.79	6.56	4.85	0.001	
Pressure(Bar)	1.161	0.537	2.16	0.053	1.23
Temperature (Degree Celsius)	0.01690	0.00580	2.91	0.014	1.44
CP_flue gas	11.80	2.10	5.61	0.000	1.03
m_flue gas	-0.000301	0.000247	-1.22	0.248	1.70

Model Summary

S	R-sq	R-sq(adj)	R-sq(pred)
2.16495	84.10%	78.31%	38.60%

Analysis of Variance

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Regression	4	272.646	68.162	14.54	0.000
Pressure(Bar)	1	21.927	21.927	4.68	0.053
Temperature (Degree Celsius)	1	39.748	39.748	8.48	0.014
CP_flue gas	1	147.392	147.392	31.45	0.000
m_flue gas	1	6.967	6.967	1.49	0.248
Error	11	51.557	4.687		
Total	15	324.204			

Fits and Diagnostics for Unusual Observations

Obs	Overall Effi (%)	Fit	Resid	Std Resid		
8	68.57	73.14	-4.57	-2.45	R	
13	69.59	68.99	0.60	1.14		X

R Large residual

X Unusual X

7. Boiler efficiency

Regression Equation

Boiler effi (%)	=	275 + 13.1 Pressure(Bar) - 0.444 Temperature (Degree Celsius) + 8.00 CP_flue gas - 0.00159 m_flue gas + 0.555 Temp diff - 2.04 Heat loss (KW)

Coefficients

Term	Coef	SE Coef	T-Value	P-Value	VIF
Constant	275	190	1.45	0.181	
Pressure(Bar)	13.1	17.1	0.77	0.463	95.36
Temperature (Degree Celsius)	-0.444	0.539	-0.82	0.431	946.33
CP_flue gas	8.00	9.12	0.88	0.403	1.48
m_flue gas	-0.00159	0.00156	-1.02	0.333	5.12
Temp diff	0.555	0.834	0.67	0.522	1040.89
Heat loss (KW)	-2.04	1.08	-1.89	0.092	25.38

Model Summary

S	R-sq	R-sq(adj)	R-sq(pred)
7.84151	83.86%	73.09%	14.03%

Analysis of Variance

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Regression	6	2874.65	479.11	7.79	0.004
Pressure(Bar)	1	36.09	36.09	0.59	0.463
Temperature (Degree Celsius)	1	41.76	41.76	0.68	0.431
CP_flue gas	1	47.28	47.28	0.77	0.403
m_flue gas	1	64.48	64.48	1.05	0.333
Temp diff	1	27.29	27.29	0.44	0.522
Heat loss (KW)	1	219.20	219.20	3.56	0.092
Error	9	553.40	61.49		
Total	15	3428.05			

Fits and diagnostics for unusual observations

Obs	Boiler effi (%)	Fit	Resid	Std Resid	
8	22.41	40.48	-18.08	-2.77	R

4.4 Model 4 - Thermodynamic analysis REG-REH with R-134a based combined power and cooling plant using vapour jet refrigeration for heat recovery of pasteurization unit of dairy industry

4.4.1 Thermodynamic Analysis

Figure 16 shows the schematic of combine REG-REH Rankine power plant

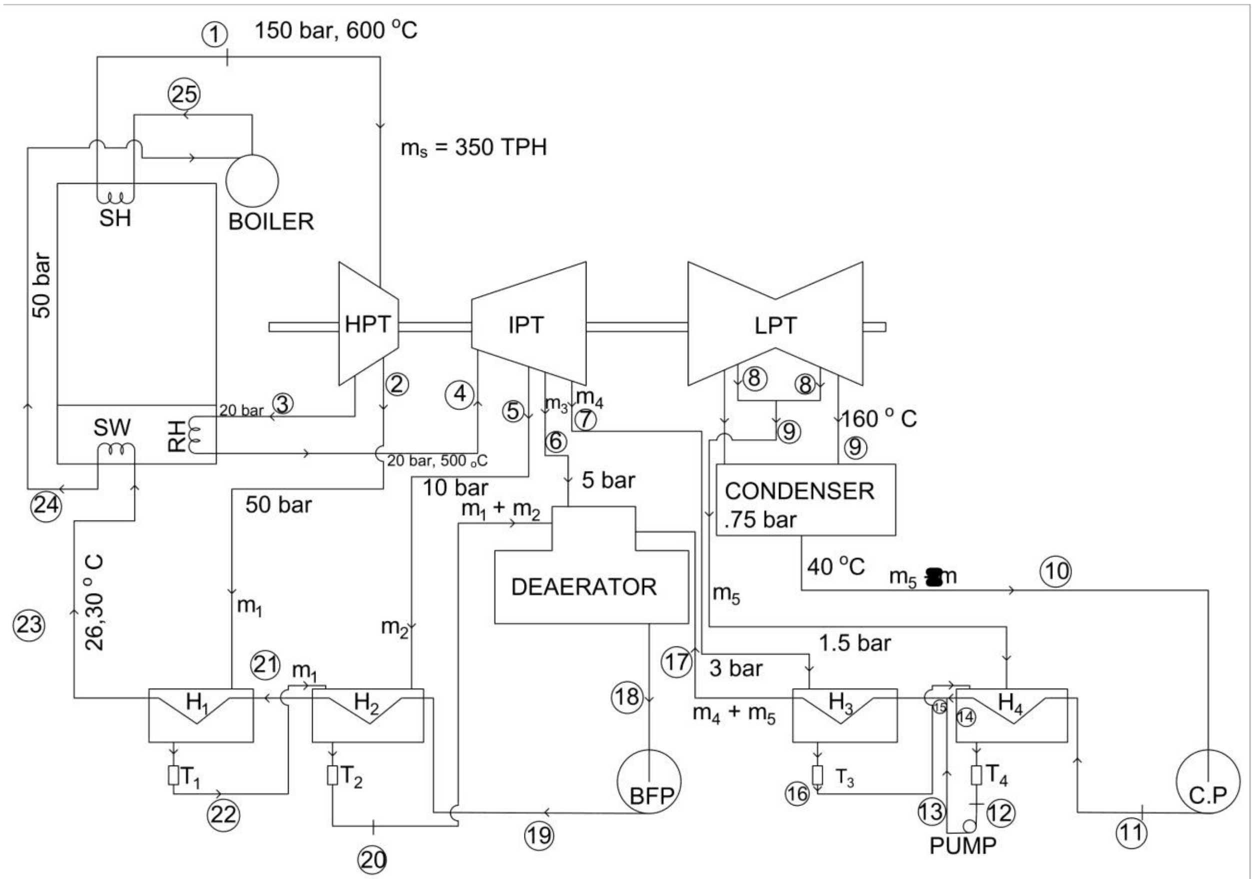


Figure 16: Combine REG-REH Rankine power plant

Regenerative Rankine cycle

110 MW plant

3T – H.P, I.P, L.P

5 Heaters – 4 closed heaters + 1 De-aerator (open heater)

$m_s = 350$ TPH

1TPH = 0.277 KG/S

H.P input – 150 bars, 600^0C

Re-heater input - 20 bars, 500^0C

At 150 bar and 600^0C

H_1= 3582.3 KJ/KG

S_1= S_2= S_3= 6.6776 KJ/KGK

At 20 bar and 300^0C

H_4= 3467.6 KJ/KG

S_4= S_5= S_6= S_7= S_8= S_9= 7.4317 KJ/KGK

T_2= 437^0 C, 50 bar, H_2= 3491 KJ/KG

T_3= 332^0 C, 20 bar, H_3= 2902 KJ/KG

T_5= 400^0 C, 10 bar, H_5= 3250 KJ/KG

T_6= 300^0 C, 5 bar, H_6= 3050 KJ/KG

T_7= 225^0 C, 3 bar, H_7= 2930 KJ/KG

T_8= 160^0 C, 1.5bar, H_8= 2790 KJ/KG

T_9= T_{10}= 40^0 C, 0.075 bar, H_9= 168 KJ/KG

For x at stage 9 (40^0 C, 0.075 bar)

S_g= S_f + xS_{fg} ………………………….……………………………………………….. (1)

Where S_{fg}= S_g - S_f…………………………………………………………………… (2)

And S_4 = S_9

From steam table S_4 = 7.431 KJ/KGK

Putting the value in equation 1

7.431 = 0.5757 + x(7.681)

x = 0.8925

$H_g = H_f + xH_{fg}$ …………………………………………………………………………… (3)

Where $H_{fg} = (H_g – H_f)$ ……………………………………………………………………… (4)

$H_9 = 168 + 0.8925\ (2406)$

$H_9 = 2315.3$ KJ/KG

$H_{11} = H_{10} +$ sp. Vol. * P ………………………………………………………………… (5)

$= 168 + 0.006 * 1.5 * 100$

$= 168.94$ KJ/KG

$T_{11} = 45^0$ C

$$\frac{T_{11}}{T_{12}} = \left(\frac{P_{11}}{P_{12}}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}$$ ……………………………………………………………………….. (6)

$T_{12} = 111^0$ C

At 1.5 bar, $H_{12} = H_{13} = H_{14} = H_{15} = 467$ KJ/KG

At 3 bar, $H_{16} = H_{17} = 561$ KJ/KG

At 5 bar, $H_{18} = 640.2$ KJ/KG

At 10 bar

$H_{19} = H_{18} +$ sp.vol. * P_{19}

$= 640.2 + 0.054*10*1000$

$H_{19} = 695$ KJ/KG

At 10 bar, $H_{20} = H_{21} = 762.8$ KJ/KG

At 50 bar, $H_{22} = H_{23} = 1154.23$ KJ/KG and $T_{23} = 263^0$c

Feed water temperature at inlet of steam generator unit is T_{23}.

Heater 1

$m_1 (H_2 – H_{22}) = m_s (H_{23} – H_{21})$ …………………………………………………….. (7)

(Assume same mass flow rate entering at the Boiler inlet from H.P turbine inlet)

$m_1 = 97\left(\frac{1154.23-762.8}{3491-1154.23}\right)$

$= 16.3$ kg/s

Heater 2

$m_2 (H_5 - H_{20}) + m_1 (H_{22} - H_{20}) = m_s (H_{21} - H_{19})$ ……………………………………….. (8)

$m_2 (3250 – 762.8) + 16.3 (1154.23 – 762.8) = 97 (762.8 – 695)$

$m_2 = 0.078$ kg/s

Heater 3 (De-aerator)

$m_3 (H_6 – H_{18}) + (m_1 + m_2) (H_{20} – H_{18}) = (m_s – m_1 – m_2 – m_3) (H_{18} – H_{17})$ ………….. (9)

$m_3 (3050 – 640.2) + (16.3+0.07) (762.8 -640.2) = (97-16.3-0.07-m_3)(640.2 -561)$

$2409.8\ m_3 + 2006.96 = (80.63 – m_3)\ 79.2$

$m_3 = 24.86$ kg/s

Heater 4

$m_4 (H_7 – H_{16}) = (m_s- m_1- m_2- m_3) (H_{17} – H_{15})$……………………………………….. (10)

$m_4 (2930 – 561) = (97-16.3-0.07-24.86) (561.19-467)$

$m_4 = 2.21$ kg/s

Heater 5

$m_5 (H_8 –H_{12}) +m_4 (H_{16} – H_{12}) = (m_s –m_1 –m_2 –m_3 –m_4 –m_5) (H_{14} –H_{11})$…………… (11)

$m_5 (2796 -467.3) +2.2 (561 -467) = (97 -16.3 -0.07 -24.86 -2.2 –m_5) (467 -168)$

$m_5 (2323) + 206.8 = (53.57 –m_5) (299)$

$m_5 = 6.029$ kg/s

Turbine Work

HPT Work = $m_s (H_1 –H_2) + (m_s –m_1) (H_2 -H_3)$……………………………………… (12)

IPT Work = HPT Work + $(m_s –m_1) (H_4 –H_5) + (m_s –m_1 –m_2) (H_5 –H_6) + (m_s –m_1 –m_2 – m_3)$

$(H_6 – H_7) + (m_s – m_1 – m_2 – m_3 – m_4) (H_7 – H_8)$……………………………. (13)

LPT Work = HPT Work + LPT Work + $(m_s – m_1 – m_2 – m_3 – m_4 – m_5) (H_8 – H_9)$……… (14)

HPT Work = 97 (3582.3 – 3491) + (97 – 16.3) (3491 – 2902)

= 8856.1 + 47532.3

= 56388.4 kJ/s

= 56 MW

IPT Work = 8856.1 + 47532.3 + 17528.04 + 16126 + 14506.2 + 7499.8

= 112042.44 kj/s

= 112 MW

LPT Work = 112042.44 + 22571.98

= 134614 KW

= 134 MW

Assume 1% of pump work

W_{net} = 132.6 MW or 132 MW

Heat Addition

$Q_1 = m_s (H_1 – H_{23}) + (m_s – m_1) (H_4 – H_3)$…………………………..……………… (15)

= 97 (3582.3 – 1154.23) + (97 – 16.3) (3467.6 – 2902)

= 235522.7 +45643.92

= 281 MW

Efficiency of cycle = W_{net}/Q……………………………………………………….. (16)

= 132/281

= 0.47

I.e. efficiency of cycle is 47%.

Temperature of water economizer (T_{24})

$\frac{T_{24}}{T_3} = \left(\frac{P_{24}}{P_3}\right)^{\frac{\gamma-1}{\gamma}}$ …………………………………………………………………………… (17)

$T_{24} = 359^0c = T_{25}$

At 150 bar, 359^0c, $H_f = H_{24} = 2373$ KJ/KG and $H_{25} = 2700$ KJ/KG

Calorific value of coal = 25200 KJ/KG

Coal consumption

$m_{coal} * C.V = Q_1$……………………………………………………………………….(18)

$m_c * 25200 = 281 * 10^3$

$m_c = 11.1$ kg/s =40 TPH

$Q_{eco} = m_s (H_{24} - H_{23})/m_c$ ………………………………………………………………… (19)

= 350 (2373 -1154)/40

= 10683.75 KJ/KG

= 10.68 Mj/kg

$Q_{Boiler} = m_s(H_{25} - H_{24})/m_c$………………………………………………………………..(20)

= 350 (2700 – 2373)/40

= 2861.25 KJ/KG

= 2.861 Mj/kg

$Q_{s.h} = m_s(H_1 - H_{25})/m_c$ ……………………………………………………………………(21)

= 350 (3582-2700)/40

= 7.71 Mj/kg

$Q_{r.h} = m_s (H_2 - H_3)/m_c$ ………………………………………………………………. (22)

= 350 (3491 – 2902)/40

= 5.15 Mj/kg

Percentage of heat absorbed by economizer = $\frac{H_{24}-H_{23}}{H_1-H_{12}}$ …………………………….. (23)

= (2373-1154)/(3582-467) =0.391

= 39.1%

Percentage of heat absorbed by Boiler = $\frac{H_{25}-H_{24}}{H_1-H_{12}}$…………………………………… (24)

= 10.7%

Percentage of heat absorbed by r.h = $\frac{H_2-H_3}{H_1-H_{12}}$ ……………………………………… (25)

= 18%

Percentage of heat absorbed by s.h = $\frac{H_1-H_{25}}{H_1-H_{12}}$ ……………………………………… (26)

= 25%

Stream rate = m_s/w_{net} ……………………………………………………………….. (27)

= 350/132

= 2.65 kg/kWh

Heat rate = Q_1/w_{net} …………………………………………………………………. (28)

= 281 * 3600/132

= 7663.6 kj/kWh

Efficiency of steam generation =$\frac{m_s(H_1-H_{23})}{m_c \times c.v}$ ……………………...………………. (29)

= 350 (3528-1154)/ 40 (25200)

= 70.6%

Work ratio = w_{net}/w_t………………………………………………………………….. (30)

= 132/134

= 98.5%

Condenser flow (condition for LPT)

= m_s - $\sum_1^5 m$

= 97 – 16.3 – 0.07 -24.86 -2.2 -6.029

= 47.5 kg/s

Water flow from C.T

$m_s(c_p\Delta t$+ L.H$)_{cond}$ = $m_w(c_p\Delta t)_{c.t\ side}$……………………………………………………(31)

47.5 (2.12 *120 + 330) = m_w (4.18*120)

M_w = 55 kg/s

For maintain 120^0 C of hot water flow at out of condenser to cooling plant

For maintain 90^0C of hot water

47.5(2.12*120 +330) = m_w (4.18*120)

M_w = 94 kg/s

We can't approach to below 90^0 C of temperature for hot water outside.

Boiler

Exergy (∈) $= m \times c_p \times T_o \left[\frac{T_b}{T_o} - 1 - ln\frac{T_b}{T_o}\right]$ ………………………………………. (32)

Where T_b = Temperature of boiler

T_o = Ambient temperature

$\in_{output} = 97.6 \times 1.996 \times 298 \times \left[\frac{873}{298} - 1 - \log\frac{873}{298}\right]$

$= 97.6 \times 1.996 \times 298 \times [2.929 - 1 - 1.0748]$

= 49.589 MW

$\in_{input}$ = 97.6 × 1.996 × 298 × $[\frac{536}{298} - 1 - \log\frac{536}{298}]$

= 97.6× 1.996 × 298× [1.7986-1- 0.587]

= 12.284 MW

$\Delta s = c_p \ln(\Delta T_{out}|\Delta T_{in})$ ……………………………………………………….… (33)

= 1.996 × ln (1.628)

= 0.9736 kj/kgk

$I_{rr} = T_o \sum \Delta s$…………………………………………………………………………. (34)

= 298×0.9736

= 290.1328 kj/kg

Rate of exergy decrease = $\epsilon_{output} - \epsilon_{input}$ …………………………………………. (35)

= 49.589 – 12.284

= 37.305 MW

Rate of exergy loss during steam generation = $\frac{\epsilon_{in}}{\epsilon_{out}}$………………………………..…… (36)

= 12.284/49.589

= 0.2477

EDD ratio = $T_0 \Delta s / Q_{Boiler}$……………………………………………………………… (37)

= 290.1477/9142.5

= 0.118

Exergetic efficiency = 1- EDD……………………………………….....………. (38)

=1- 0.118

= 0.882

Rate of exergy increase in stream = $m_s [H_1 - H_{23} - T_0(s_1 - s_{23})]$…………….…………..(39)

=97.6 [3582.3 -1154.23 -298(6.6776 -2.2835)]

= 109.178 MW

Exergy destruction in Boiler = rate of exergy increases in steam – rate of exergy decrease………………………………………………………………………. (40)

= 109.178 – 37.305

= 71.873 MW

Condenser

m= 47.5 kg/s

Exergy (∈) $= m \times c_p \times T_o \left[\frac{T_b}{T_o} - 1 - ln\frac{T_b}{T_o}\right]$

$\in_{input} = 47.5 \times 1.996 \times 298 \times \left[\frac{433}{298} - 1 - \log\frac{433}{298}\right]$

= 47.5 × 1.996 × 298 × [1.45 -1- 0.3738]

= 2.157 MW

$\in_{output}$ = 47.5 × 1.996 × 298 × $[\frac{318}{298} - 1 - \log\frac{318}{298}]$

= 0.048 MW

$\Delta s = c_p \ln(\Delta T_{out} | \Delta T_{in})$

= 1.996 ln (433/313)

= 0.6477 kj/kgk

$I_{rr} = T_o \sum \Delta s$

= 298× 0.6477

= 193.0146 kj/kg

Rate of exergy decrease = $\in_{output} - \in_{input}$

= 2.157 – 0.048

= 2.109 MW

EDD ratio = $T_0 \Delta s / Q_{Boiler}$

= 193.0146/2147.3

= 0.0898

Exergetic efficiency = 1-EDD

=1- 0.0898

= 0.9102

Rate of exergy increase in stream = $m_s [H_9 - H_{10} - T_0(s_1 - s_9)]$

=47.5 [2315.9 -168 -298(7.431 -2.562)]

= 33.104 MW

Exergy destruction in condenser = rate of exergy increases in steam – the rate of exergy decrease

= 33.104 – 2.109

= 30.995 MW

Thermodynamic Analysis of integrated VJR

Combined cooling and power plant VJR are shown in figure 17.

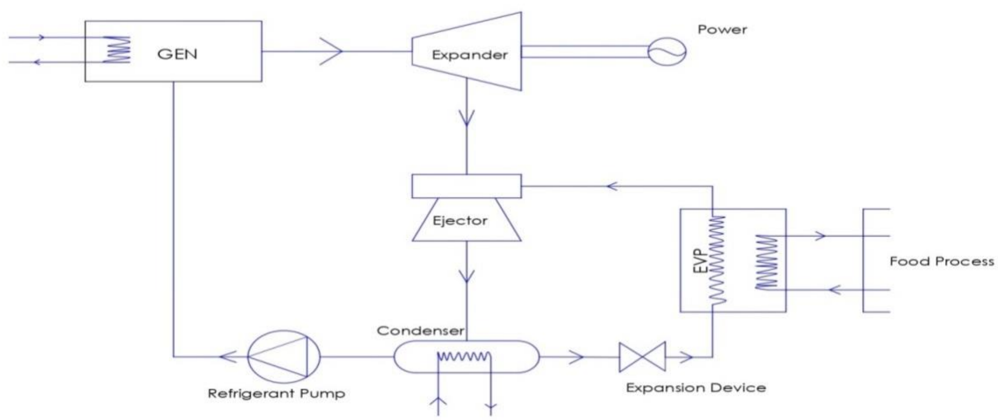


Figure 17: Combined cooling and power plant VJR

Assume

Cooling capacity = 35 KW or 10 TOR (ton of refrigeration)

Nozzle Efficiency-n_{ozzle} = 90%

Entrain Efficiency-η_{ent} = 70%

Thermo-Compressor Efficiency-η_{comp} = 80%

Turbine Efficiency-$\eta_{turbine}$ = 90%

The mass and energy equation are considered for thermodynamic analysis of VJR-

$$\eta_{nozzle} = \frac{h2' - h3\prime}{h2' - h3} = \frac{Actual\ Enthalpy\ Drop}{Isentropic\ enthalpy\ drop} \quad \ldots\ldots (1)$$

$$\eta_{expansion} = \frac{h1 - h2\prime}{h1 - h2} = \frac{Actual\ enthalpy\ drop}{Isentropic\ enthalpy\ drop} \quad \ldots\ldots (2)$$

As per data of pasteurization plant

P1= 35 to 45 Bar, T1= 90-125 Degree Celsius

(at 35 bar and 90 Degree C)

From steam table, h1= 420 KJ/Kg, s1=1.649 KJ/KgK

P2=P2'=7 Bar, T2=28 Degree C

From steam table- h2=413.8 KJ/Kg, hf2=238.8 KJ/Kg, sf2=1.134 KJ/KgK, s2= 1.715 KJ/KgK

From equ (2) h2' = 413.7 KJ/Kg

P3=P3'=P5=P6=P4=2 BAR=Evaporator (P_E)

hf3=hf3'=186.7 KJ/Kg, sf3=0.94 KJ/Kg-K, s3=1.734 KJ/Kg-k,hg3=hg3'=392.06KJ/Kg

s2= s3 =sf3+ x3 sfg3 = sf3+ x3(s3-sf3)

And get x3 = 0.97

h3= hf3 +x3 hf3 = hf3 + x3 (hg3-hf3)

h3= 385.89KJ/Kg

$$\eta_{nozzle} = \frac{h2' - h3'}{h2' - h3}$$

h3' = 388.63 KJ/Kg

h3'=hf3' +x3' hf3' = hf3' + x3' (h3'-hf3')

Put all values and get x3'=0.983 (which is more than x3)

The steam is coming out from nozzle at very high velocity at point 4 entrance and resulting condition of mixture is representing by 4, at linePE = 2bar and dryness fraction=0.97

$$\eta_{entrain} = 0.70 = \frac{h2 - h4}{h2' - h3'}$$

Put all values and get h4=396.15 KJ/Kg

From steam properties – sf4=0.9804 KJ/Kg-K, sg4=1.729 KJ/Kg-K

At P5 =P_C= 10 bar

h5=419.43 KJ/Kg, hf3=256.6 KJ/Kg, sf5 = 1.190 KJ/Kg-K, sg5=s5 = 1.711 KJ/Kg-K

s4=s5 = sf4 + x4 (sg4-sf4)

Andat x4= 0.97

Thermo-Compressor Efficiency=η_{comp} = 80% = (h5-h4)/(h5'-h4)

H5'=425.75 KJ/Kg

Mass of motive steam and vapour refrigerant in evaporator

$$\frac{m\_s}{m\_v} = \frac{(h5 - h4)}{\left[(h2' - h3) \times \eta_{nozzle} \times \eta_{entrain} \times \eta_{comp} - (h5 - h4)\right]}$$

m_v or m_ref = mass of vapour refrigerant

Put all values

m_s/m_v = 1.4 kg /kg of flash vapour

Mass of refrigerant-

Refrigeration Effect= RE = h3-Cp_ref (-10 degreeC) at 2 bar

RE=385.89-1.316 KJ/Kg-K (263K)

35KW =39.782 KJ/Kg of refrigerants

m_ref =1.13 kg/s

m_s/m_v = 1.4 , so m_s = 1.13×1.4 = 1.57 kg/s

Generator Heat = Q_gen = m_s(h1-h2)

Q_gen= 1.57 (420-413.8) = 10 KW

COP_th =$\frac{RE}{Q\_thermalcompressor} = \frac{RE}{[h5'(m_s + m_{ref}) - m_s \times h2' - m\_ref \times h5']}$

Put all values and get COP = 0.538 [range of COP of VJR varies from 0.5 to 0.65]

COP_carnot = $\frac{T_{evp}(T\_gen - T_{cond})}{T\_gen(T_{cond} - T_{evp})}$

COP_carnot = $\frac{263(90-40)}{363(40-(10))}$ = 0.72

COP_exergy = $\frac{COP\_th}{COP\_carnot}$= 0.73

CHAPTER 5

Chapter-5 Result and discussion

The present chapter is discussing the result of different thermodynamic systems which are investigated in this research work. The thermodynamic analysis has been done with mass-energy balance equation, entropy generation approach with DOE application. The several performance parameters have examined and provide valuable outcomes. The graphical and tabular form of results have mentioned model-wise as followings.

5.1 Result analysis of condenser heat recovery of coal fired power plant by ORC

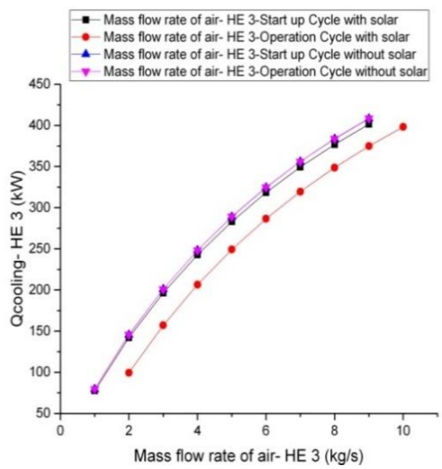


Figure 18: Effect of mass flow rate of air on cooling in HE3

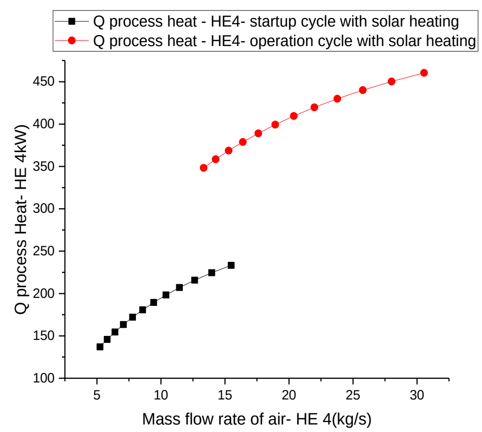
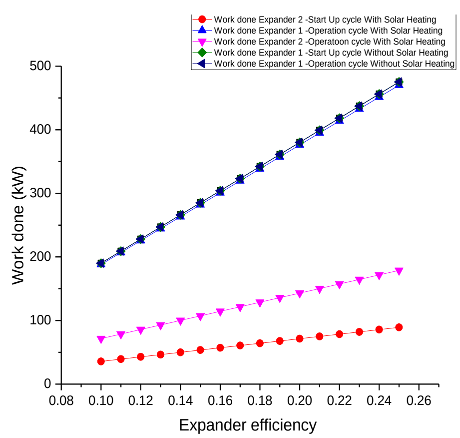


Figure 19: Effect of mass flow rate of air on process heat for HE 4



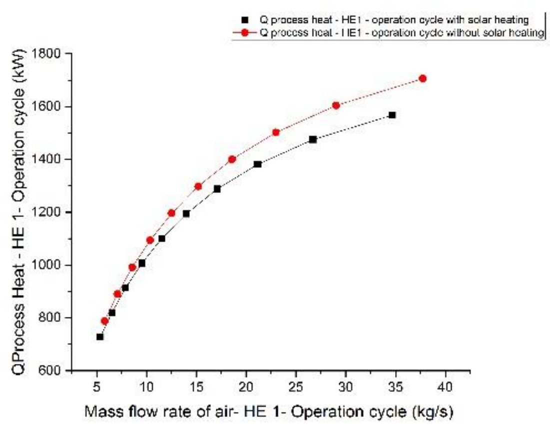


Figure 20: Effect of expander efficiency on expander work

Figure 21: Effect of mass flow rate of air, flowing through HE 1, on Q process heat in operation cycle and startup cycle

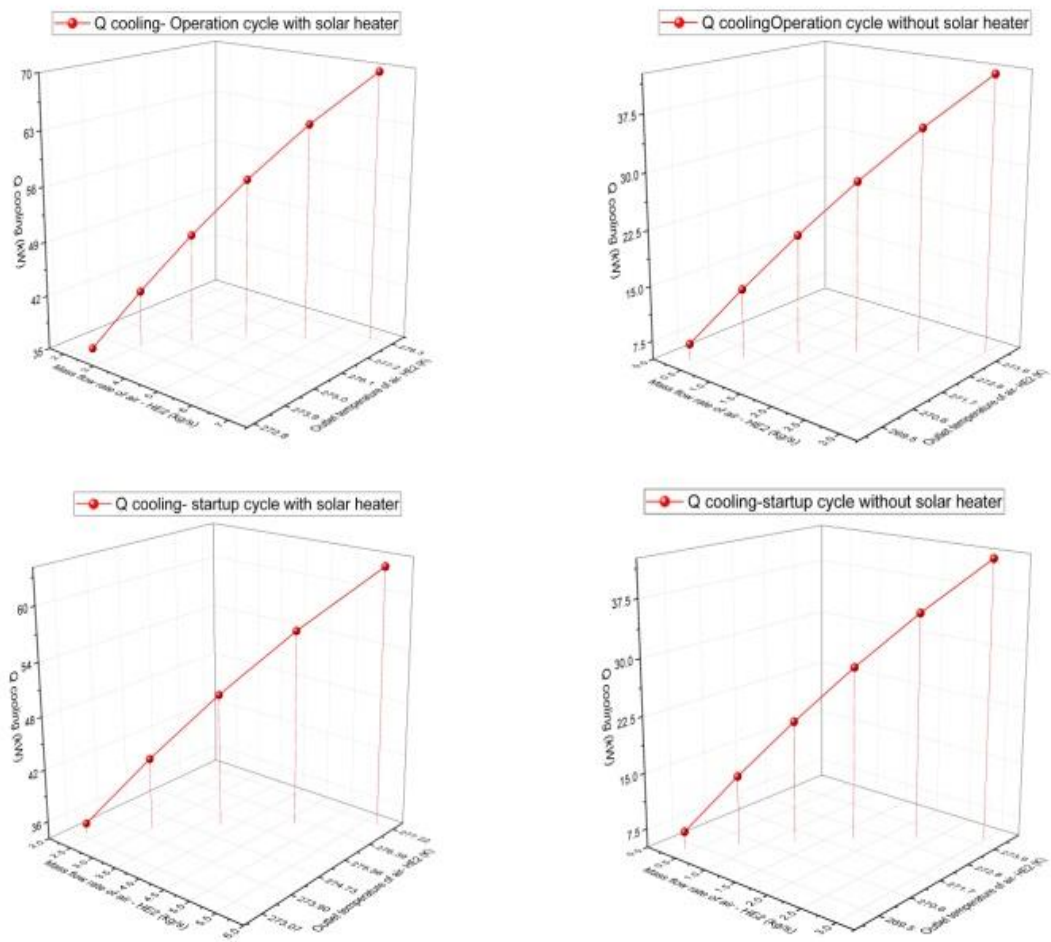


Figure 22: Qcooling Vs. mass flow rate of air- HE 2Vs Air outlet temperature – HE 2

Fig 18 and Fig 22: Increasing of mass flow rate of air increases the rate of heat exchange, thereby raising the temperature of the cold refrigerant more rapidly per unit time. This increase the amount of heat rejected by air or Qcooling but has a negative effect on the decrement of the air temperature leading to an increase being observed in the temperature of air at the outlet of HE 2 due to upward shift of the equilibrium point of heat exchange. By the Increasing the mass flow rate of air in HE2 increases the heat obtained thereby reducing the cooling effect and heating the air more. In operation cycle with solar heater the change in heat obtained and the rise in air temperature for a small increase in mass flow rate of air is much more than that in the operation cycle without solar heater is explain in fig-19.

Fig 19: Integrating solar heating with operation cycle increases the process heat obtained for industrial purpose. Fig 20: The efficiency of scroll type expander is in the range of 14% to 25%. In case of solar integration more heat for industrial process will be obtained. Fig 21: Before solar heating, heat process will gain more by heat exchanger-1 in both cycle, after solar heating process heat improve slowly in operational cycle. The fig-23 explains the additional heating through solar heating provide more expander work and as well as thermal efficiency also achived. Only single expander work is achieved witgout solar operation cycle, but solar thermal system provides more heaing and double stage of expander work. The more solar heating also causes of heat loss,and it also decreses the thermal efficiency. The additional heating through solar aopplicable for when main plant discharged heat availability is less.

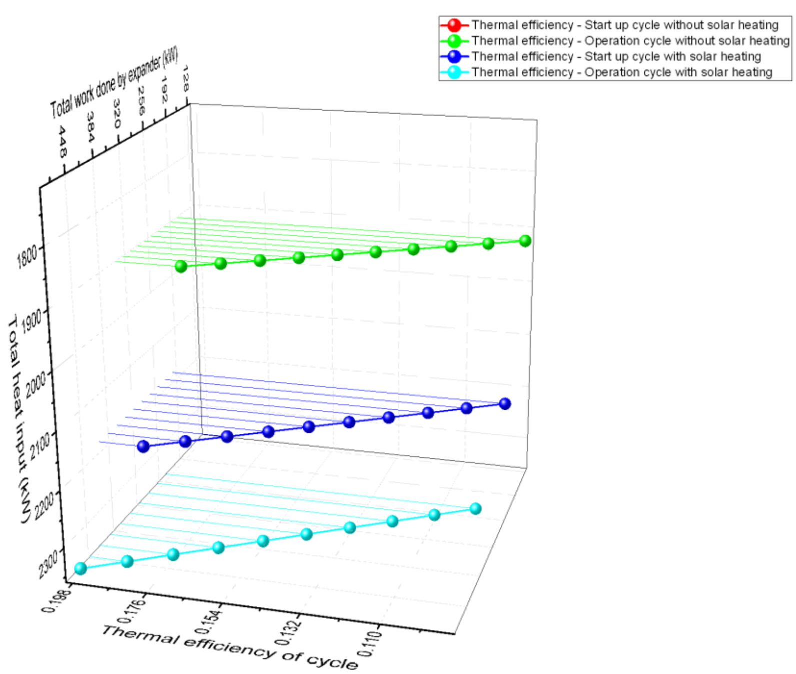


Figure 23: Total work done by expander Vs. Total heat input Vs. Thermal efficiency of cycle

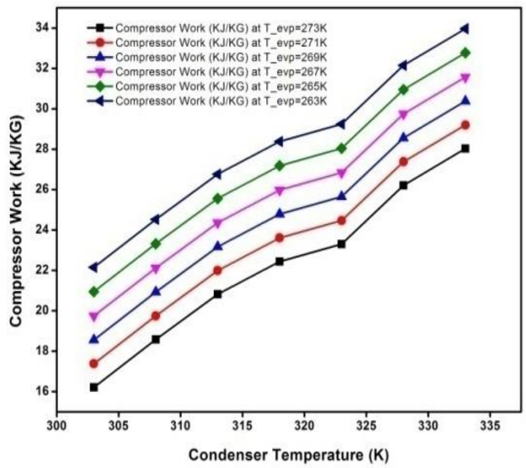


Figure 24: Effect of T_condenser in compression work

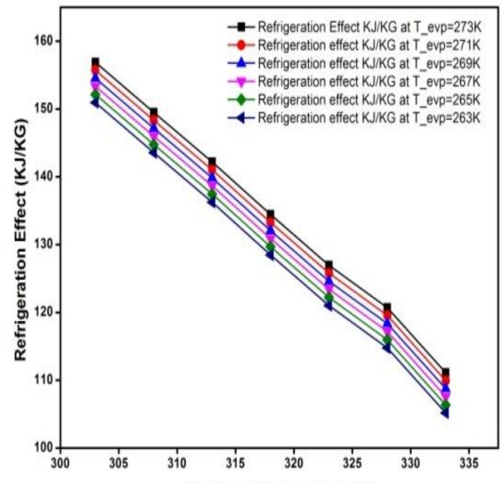


Figure 25: Effect of T_condenser in refrigeration effect

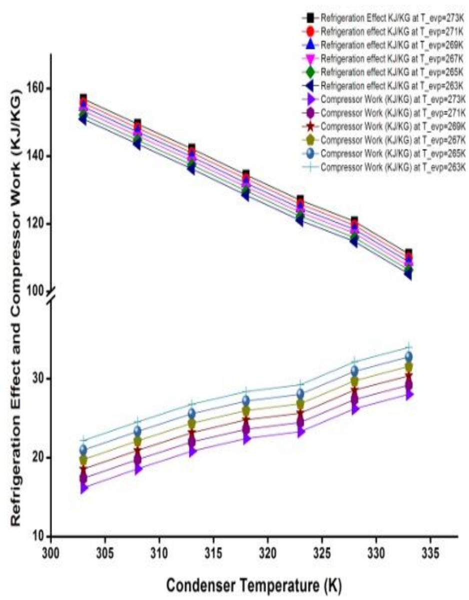


Figure 26: Effect of T_Condenser in Compression Work

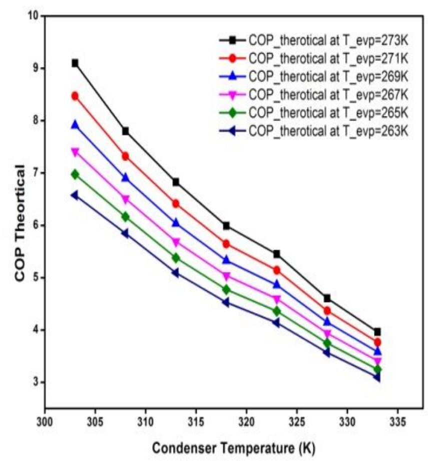


Figure 27: Effect of T_Condenser theoretical COP

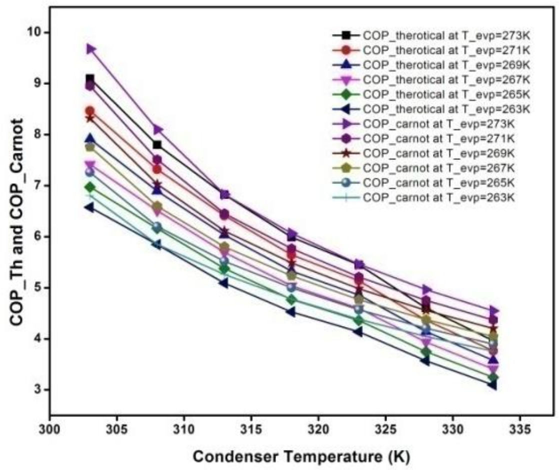


Figure 28: Effect of T_Condenser in COP_I & COP_II

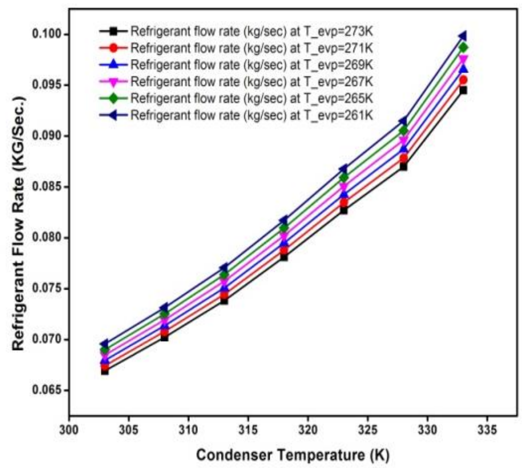


Figure 29: Effect of refrigerant mass flow rate with T_evaporation

The compressor of VCRS is consuming more power when it operates for higher condenser temperature. If compressor work is more, then cooling effect will decreases and further COP cannot be achieved. Fig 24, 25 &26 explains the effect of higher rate of heat rejection through condenser on compression work and COP with different evaporation temperature. The Carnot COP is slightly above the theoretical COP of VCRS, The refrigerant flow rate also effectively more when it operates for low evaporator temperature. The operating pressure at condenser line is more, this pressure difference causes the flow rate of refrigerants. COP and mass flow rate variation with evaporation and condenser temperature is explained in fig 27, 28 &29. Table-10 discusses the effect of operating temperature for both. When higher temperature of VCRS system is increase the COP is decreasing due to excess superheating of refrigerant but VARS gives improved COP when generator temperature (higher temperature of VARS) is more.

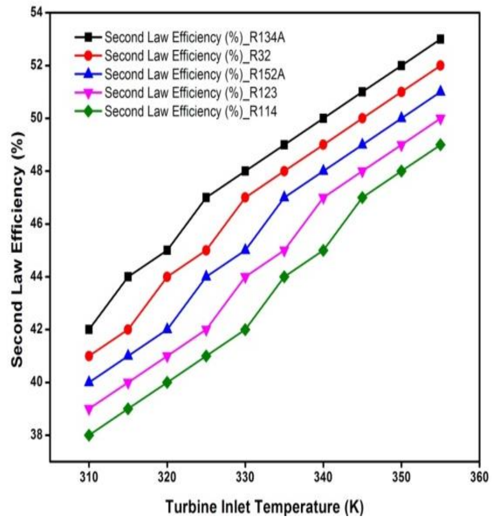


Figure 30: Effect of TIT on 2nd law efficiency with different eco-friendly refrigerant

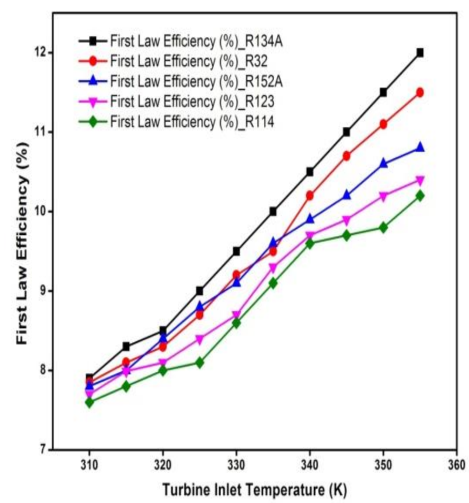


Figure 31: Effect of TIT on 1st law efficiency with different eco-friendly refrigerant

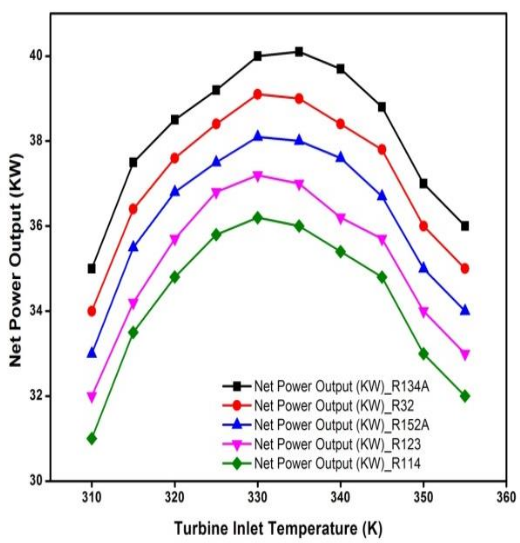


Figure 32: Effect of TIT on ORC net output with different eco-friendly refrigerant

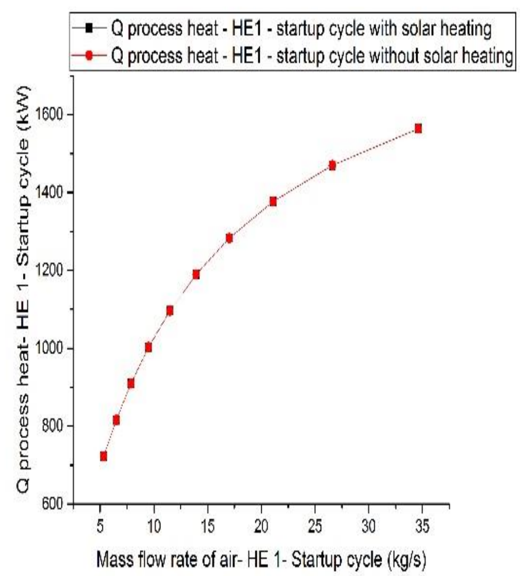


Figure 33: Effect of mass flow rate of air, flowing through HE 1, on Q process heat in operation cycle and startup cycle

The R134a refrigerants perfrom better in terms of 1st and 2nd law efficiency of ORC system with higher turbine inlet temperature (TIT) which is shown in figure 30 & 31 but it will decrease due to thermal loss and more superheating of refrigerants at higher TIT, the ORC output is decreases with higher TIT is explain in figure-32. The solar operated heat recovery system have better perfromance as shown in table-10. Effect of mass flow rate of air, flowing through HE 1, on Q process heat in operation cycle and startup cycle is depicted by figure 33.

Table 10: Parametric result of ORC heat recovery system

Parametric Results	**Cycle without solar**	**Cycle with solar**
$\eta_{thermal}$	8-15%	12-27%
$\eta_{expander}$	10-25%	10-25%
Process heat ($Q_{process\ heat}$)	900-1100 kW	1200-1500 kW
Cooling effect ($Q_{Cooling}$)	14- 19.25 kW	35- 38.5 kW
Expander work ($W_{expander}$)	200-470 kW	260-650 kW
Solar heating (Q_{Solar})	N/A	460 kW
Solar collector efficiency	N/A	20-25%
Water temp through solar field	N/A	343.15-363.15 K

5.2 Result analysis of stack flow heat recovery using LiBr-H2O VARS of combined GT-ST power generation system.

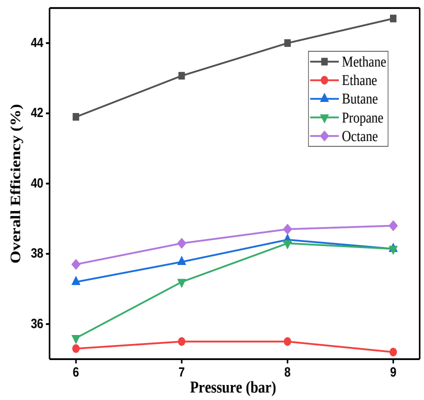


Figure 34: Effect of operating pressure on GT plant overall efficiency with different gas combustion

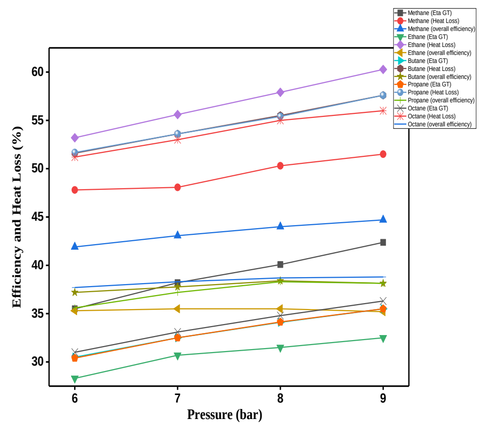


Figure 35: Effect of operating pressure on GT plant performance with different gas combustion

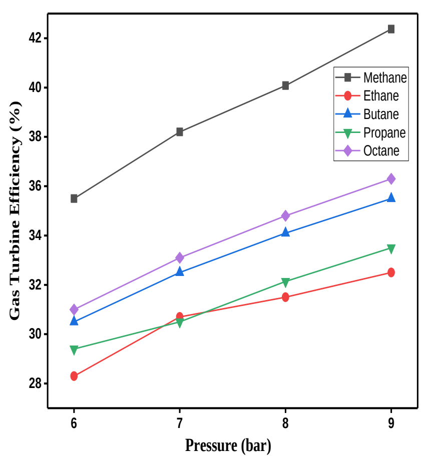


Figure 36: Effect of operating pressure on GT plant gas turbine efficiency with different gas combustion

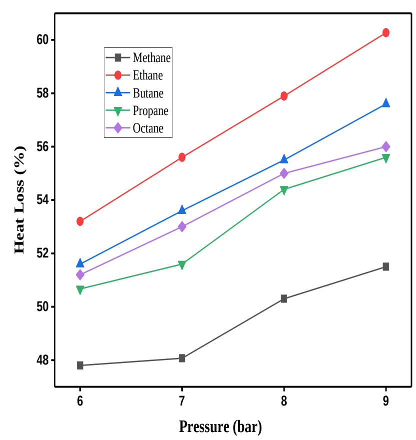


Figure 37: Effect of operating pressure on heat loss of GT plant with different gas combustion

The more compression ratio enhances the GTIT and higher temperature of GT inlet gives higher thermal efficiency. Fig 34 to 37 are showing the effect of compression ratio in overall efficiency of combined GT-ST plant with five different gas combustion in GT plant.

The maximum GTIT taken 1000^0C, the metallurgical criteria of turbine material is main factor for limiting value of GTIT. Methane gas have higher efficiency with high CR and GTIT values, fig-36 clearly indicate minimum heat loss also, where as other gases have valuable output with all ranges of CR values.

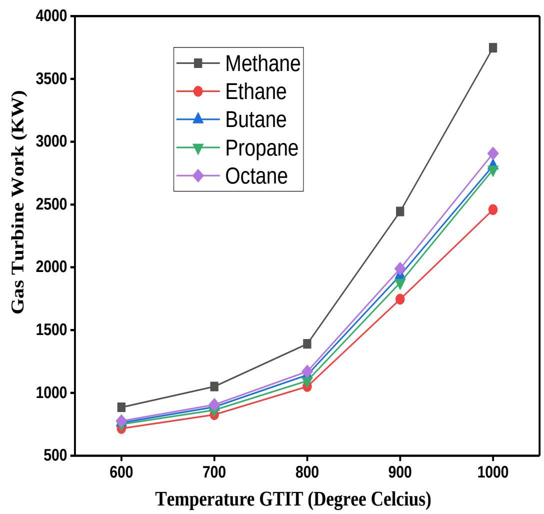


Figure 38: Effect of GTIT on gas turbine work

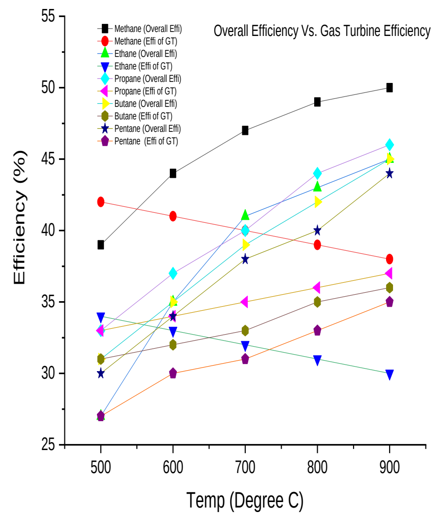


Figure 39: Effect of GTIT on gas turbine efficiency and overall efficiency

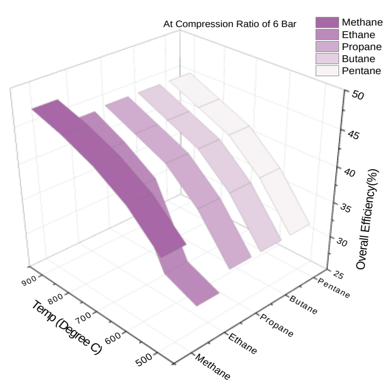


Figure 40: η_overall at 6 Bar of pressure ratio

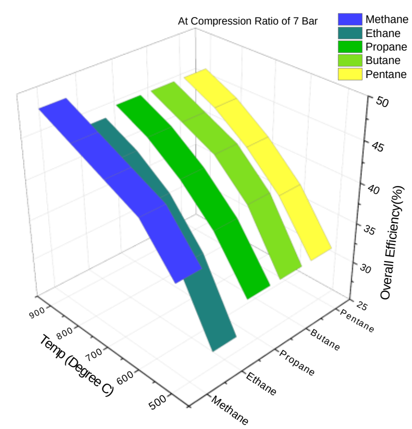


Figure 41: η_overall at 7 bar of pressure ratio

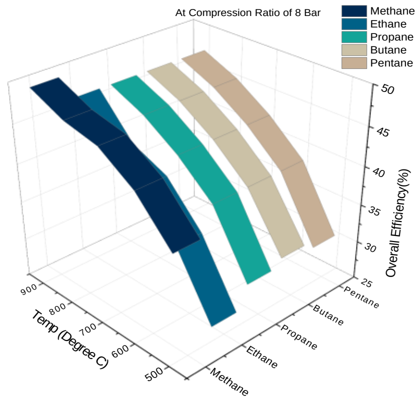


Figure 42: η_overall at 8 Bar of pressure ratio

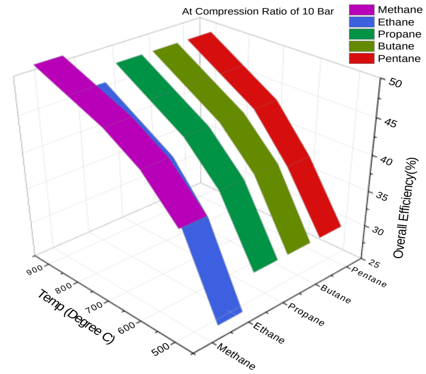


Figure 43: η_overall at 9 Bar of pressure ratio

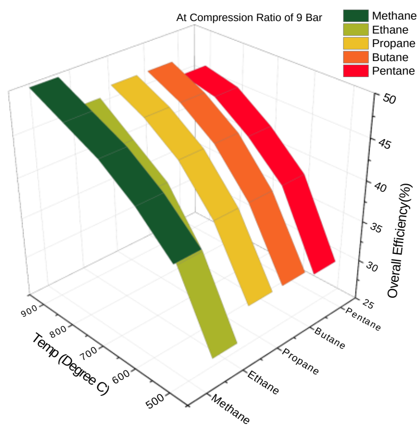


Figure 44: η_overall at 10 Bar of pressure ratio

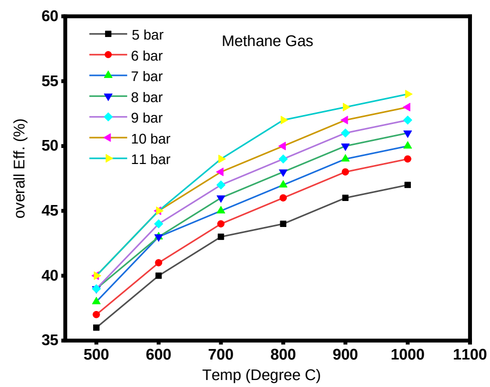


Figure 45: η_overall at different GTIT& pressure for methane combustion

The Effect of GTIT on gas turbine work and Effect of GTIT on gas turbine efficiency & overall efficiency are clearly explained in figure 38 and 39 respectively.

The compression ratio helps to enhance the gas turbine inlet temperature, and higher temperature of GT inlet gives higher thermal efficiency. Fig 40-44 are showing the effect of compression ratio in overall efficiency of combined GT-ST plant with 5 different gas of burning in combustion chamber of GT plant. Methane gas have always higher efficiency with high CR and temp values,where as other gases have valuable output with all ranges of CR values. The performance also influenced by the heat loss during the coupling of GT-ST cycle. In all pressure and GTIT value methane gas perform better. Fig-45 explains at higher pressure of 10-11 bars and higher GTIT methane gas gives maximum efficiency. The higher GTIT discharge more gas turbine exhaust temperature which is input for ST plant operation.

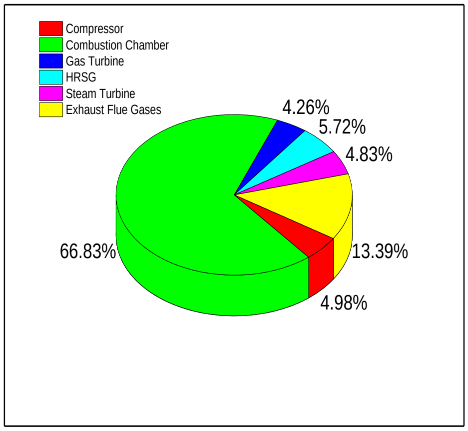


Figure 46: Exergy loss in GT-ST plant components

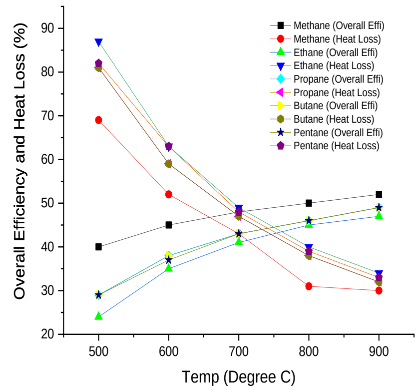


Figure 47: Effect of GTIT on overall efficiency and heat loss

The fig-46 shows the maximum irreversibility (energy loss) found in plant components. The CC of GT plant has maximum energy loss due to massive heat transfer from CC during the combustion of fuel. Exhaust flue gas unit has a significant energy loss due to high temperature reduction during the stack flow and possibilities of dew formation of pollutants of flue gases. The irreversibility of steam turbine is more than the gas turbine because in steam turbine expansion phase change of steam is start and gets to condense, but no phase change is possible in gas turbine expansion.

The overall plant output is greatly affected by the heat loss.The maximum possible GT efficiency is achieved as 42% at 900 Degree C with methane gas firing as shown in fig-47 .The heat supply through combustion chamber is converted in to GT workdone and GT cycle heat rejection further, but rejected heat of GT cycle is utilize through HRSG of steam power plant for ST work output. If heat loss is more, than overall performance will decreases as shown in fig-48. At low temperature of gas turbine inlet temperature heat loss is more but at higher temperature rejected heat is completely utilize and minimum heat loss enhance the overall performance.

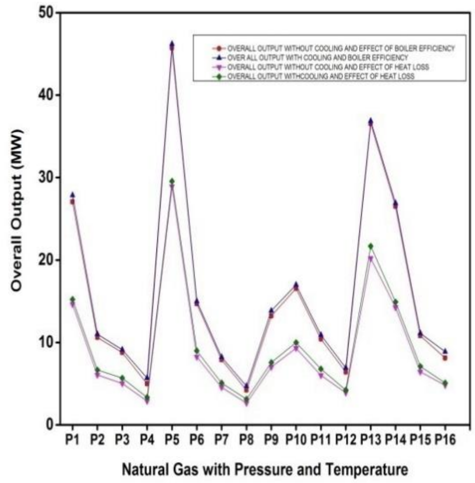


Figure 48: Effect of combustible gas pressure and temperature on output

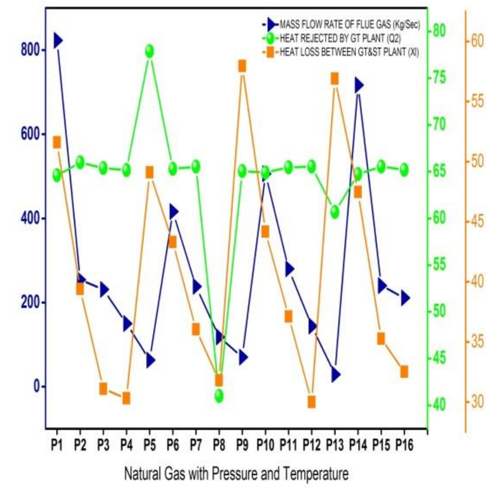
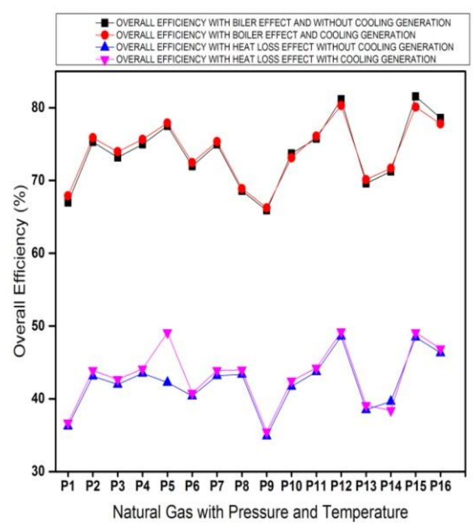


Figure 49: Effect of GTIT on overall efficiency and heat loss



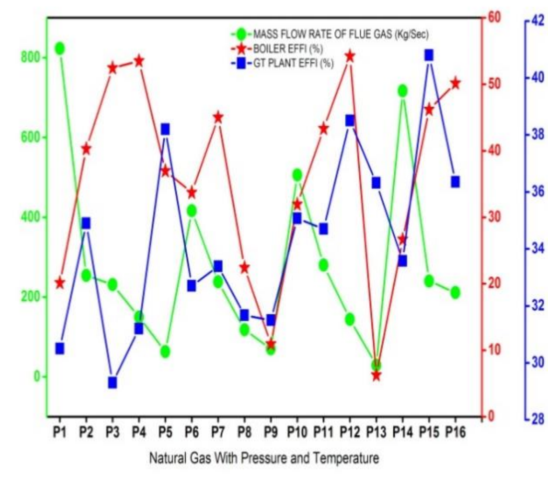


Figure 50: Effect of overall efficiency with cooling and without cooling

Figure 51: Effect of combustible gas pressure and temperature on overall efficiency, GT performance and heat loss

The effect of GT plant heat rejection, mass flow rate of flue gas and heat loss between GT&ST plant contributes majorly for plant function and operation. Fig 48-51 have been explained the all above discussed factors for plant output and performance. If mass flow rate of flue gas is more heat rejection from GT plant is more and this rejected heat is useful heat for ST plant operation, so GT plant efficiency is less as compare to boiler efficiency with less heat loss during the combining of both power generation cycle in many set of compressor pressure, GTIT and natural gas (P1,P2,,,,,,,,P16).The all output results in terms of efficiency and plant work done also consider the cooling and without cooling effect, because the integration of small unit of VARS is utilize the stack flow heat of plant and the exit temperature of flue gas must be maintain about 150 0C and in this analysis the stack flow temperature is estimated about 165^{0}C, which is valuable heat source for VARS generator.

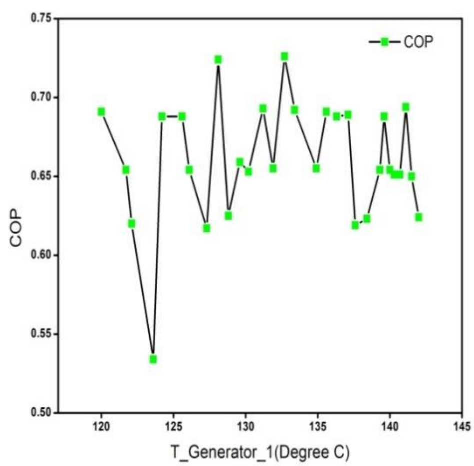


Figure 52: COP of VARS and Generator_1 temperature

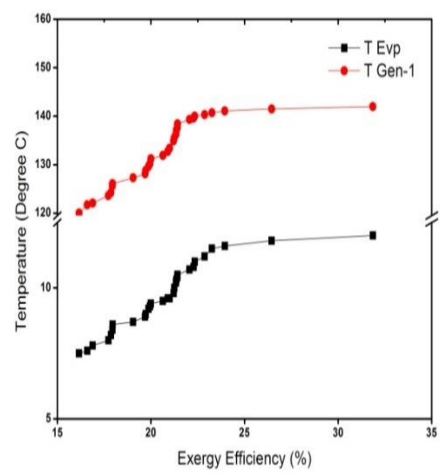


Figure 53: Effect of evaporator and generator temperature on exergy

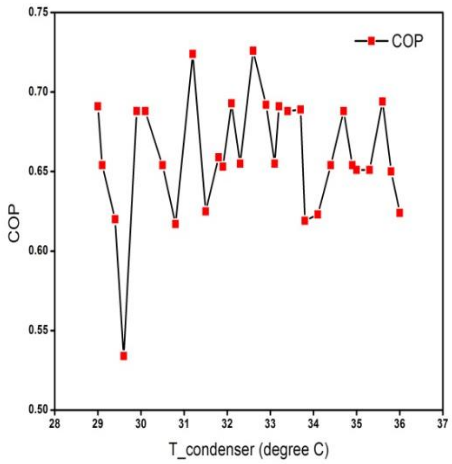


Figure 54: COP variation with the effect of condenser temperature

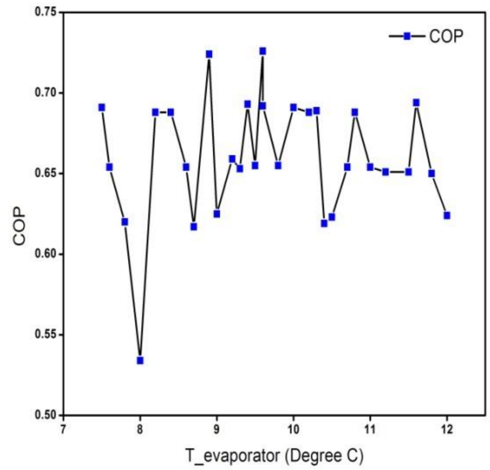


Figure 55: COP variation with the effect of evaporator temperature

The effect of operating parameter like generator, condenser and evaporator temperature on performance of VARS system gives significant output. The higher generator does not give higher COP because heat loss will start. The moderate value of generator temperature (1200C-1400C) gives remarkable COP (0.70-0.75) of proposed VARS as shown in fig-52. Similarly, the average value of condenser temperature (30 ^{0}C-34 ^{0}C) and 80C -100C of evaporator temperature gives valuable COP as shown in fig- 54 & 55. Observation Tables of GT-ST Plant performance (operated with octane gas) is shown in table 11.

Table 11: Observation Tables of GT-ST Plant performance (operated with octane gas

Performance Parameters	**Parametric Values**	**Performance Parameters**	**Parametric Values**	**Performance Parameters**	**Parametric Values**
$W_{\_STplant}$	29MW	Gas Turbine Exergy Destruction	5646 KW	Exhaust Flue Gases Exergy Destruction	17760 KW
$W_{\_GT\ plant}$	53MW	HRSG Exergy Destruction	7583 KW	Total I_{LOSS} Or Destruction $=\sum Iloss=$	132945 KW
$\eta_{STplant}$ & $\eta_{GTplant}$& $\eta_{overallplant}$	38% & 27% & 41.7%	Steam Turbine Exergy Destruction	6412 KW	Exergy Input (ΔG_0)	212810 kW
Heat lost during combining of GT-ST cycle $=X_L$	35.4%	Compressor Exergy Destruction	6613 KW	Total Output	82,000KW
m_gas / m_air /m_fuel	275.4Kg/s /271.3kg/s /4.46 Kg	Combustion chamber Exergy Destruction	88661 KW	Overall Exergetic Efficiency$=\eta_{EX}$	**38.5%**

m_gas / m_air /m_fuel	275.4Kg/s /271.3kg/s /4.46 Kg	Combustion chamber Exergy Destruction	88661 KW	Overall Exergetic Efficiency=η_{EX}	**38.5%**
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$\eta_{STplant}$ & $\eta_{GTplant}$& $\eta_{overallplant}$	38% & 27% & 41.7%	Steam Turbine Exergy Destruction	6412 KW	Exergy Input (ΔG_0)	212810 kW
Heat lost during combining of GT-ST cycle $=X_L$	35.4%	Compressor Exergy Destruction	6613 KW	Total Output	82,000KW

The fig-53 explains the exergetic performance of VARS, which is achieved with 1200C-1400C of generator temperature.

The higher side temperature value of generator and condenser not suitable for high COP. The recommended COP of LiBr-H2O based VRAS is 0.60 to 0.75.

Design of experience analysis Results-MLR Method

Regression Equation Overall Effi (%)

$$= 46.6 + 0.423\ \text{Pressure(Bar)} - 0.01883\ \text{Temperature (Degree Celsius)} - 0.798\ \text{CP\_flue gas} + 0.000009\ \text{m\_flue gas} + 0.0260\ \text{Temp Diff} - 0.0140\ \text{Topping Plant Heat Rejection} - 0.002\ \text{Heat Loss (KW)} + 0.2270\ \text{BOILER Effi (\%)} + 0.9367\ \text{Topping plant Effi (\%)} - 0.090\ \text{Bottoming plant effi (\%)}$$

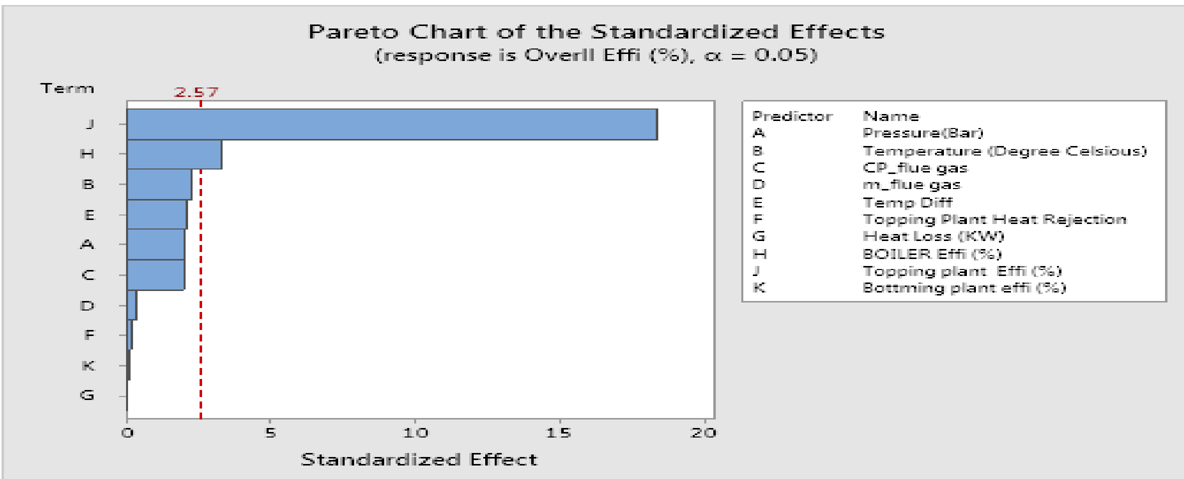

Figure 56: Pareto chart of overall efficiency performance indicators

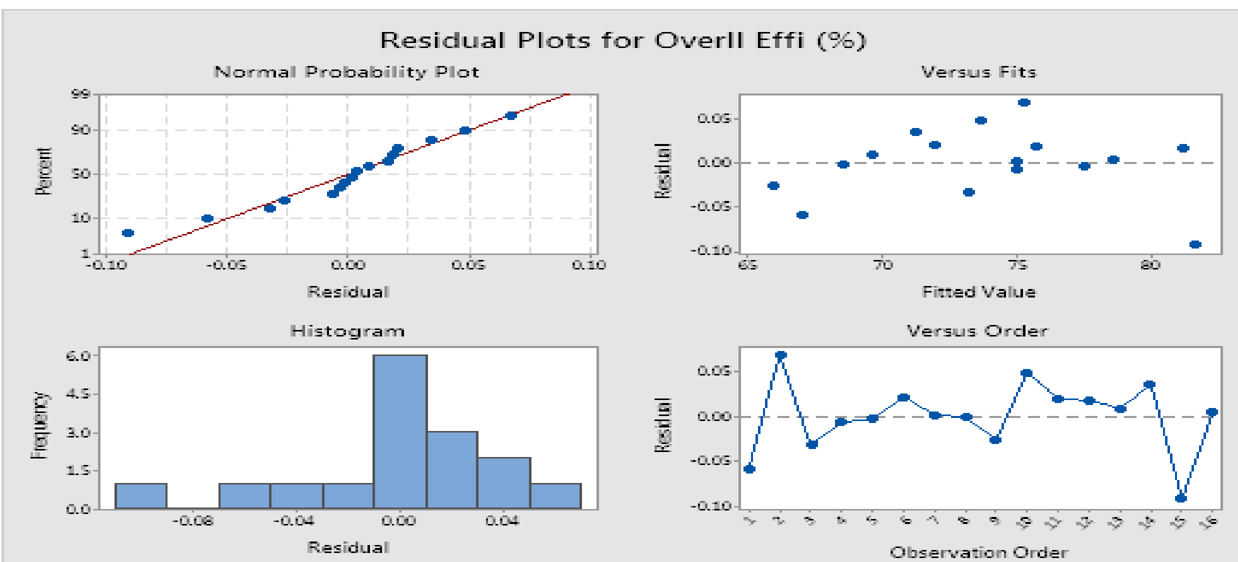

Figure 57: Residual plot of overall efficiency performance indicators

The MLR model summary measures the overall efficiency of the plant. The adjusted R-squared value of 64.92% and the Predict R-squared value of 51.99 % are fairly acceptable for this analysis. Furthermore, a value of R2 on a higher side indicates that the results are close to the real data values. A comparison between the actual data and the predicted one is highlighted in Table

12. A set of data based on 16 runs were collected as per the operating gas turbine pressure, GTIT and combustible natural gases. The table was constructed which included the predicted responses from MLR analysis. The values seem to coincide with the actual data which is considered good.

It also shows the residual by response surface and the percentage variation versus the run number in fig-57. A pareto plot of residuals is shown in Figure-56. The graphs highlight one key fact, that is, the value of R2 is 71.93%, shows minimum irregularity. The linear model is statistically adequate. It can be seen that the best-fit line resembles the (Y=X) line when plotted on a data set of 16 points. This indicates that the predicted values and the real data are in close proximity to each other. The variation found in actual results from the predicted ones was between 2.612 and -

5.386 showing only a minor dissimilarity.

Regression equation of Topping Plant Heat Rejection

= 206 + 13.4 Pressure(Bar) - 0.441 Temperature (Degree Celsius)
+ 9.68 CP_flue gas - 0.00163 m_flue gas + 0.611 Temp Diff

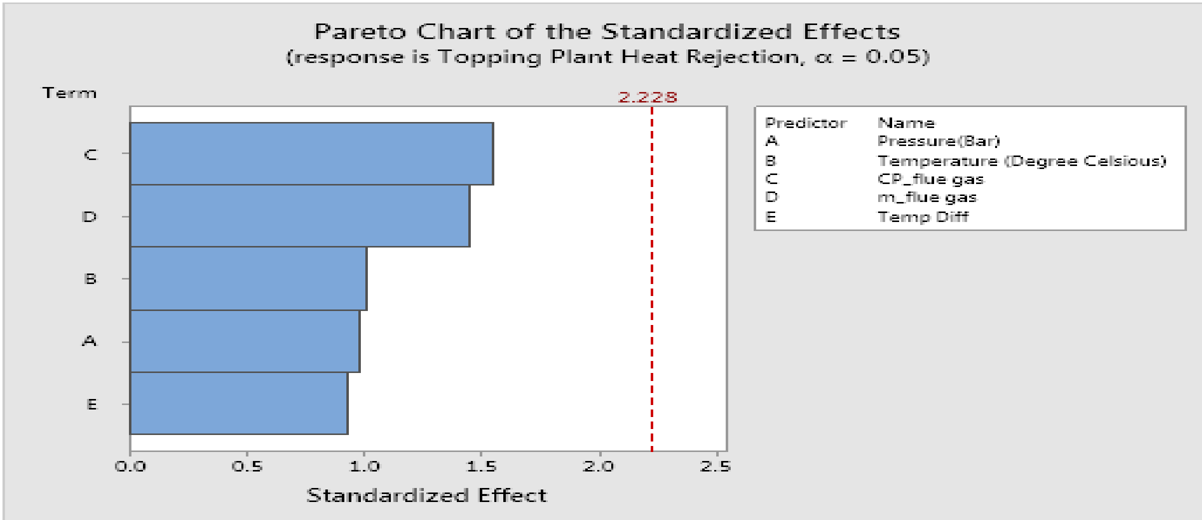

Figure 58: Pareto chart of topping plant performance

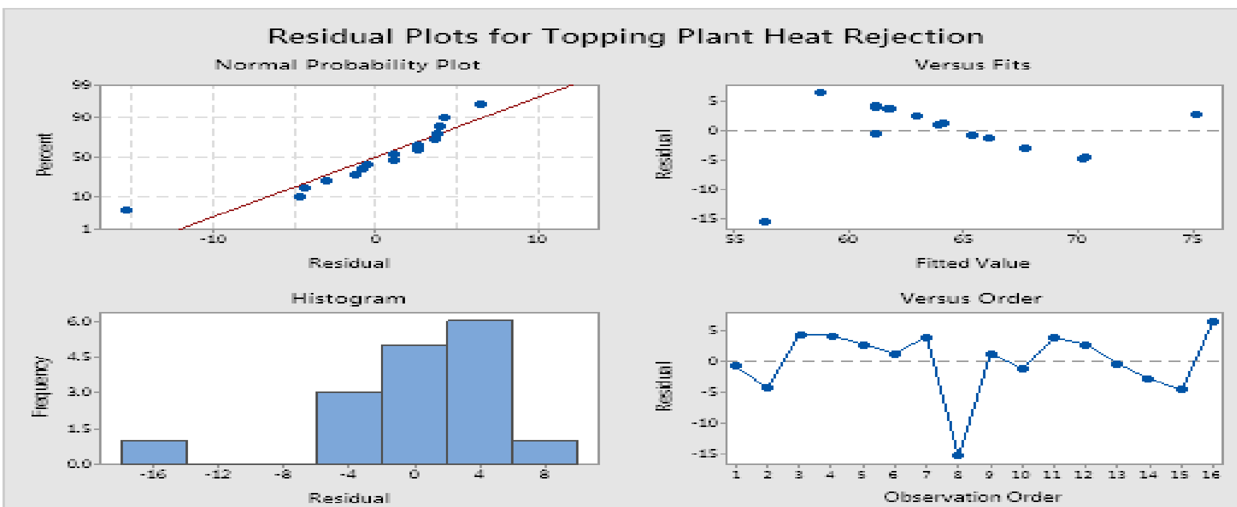

Figure 59: Residual plot of topping plant performance

Fig 58 & 59 are explaining the residual plot for the topping plant efficiency. The residual line show no deviation of test run, and pareto charts show the pressure, mass flow rate and cp of flue gases are considerable variable for the topping plant performance.

Regression Equation-Overall Effi (%) considering of GT-ST cycle heat loss, boiler GT and ST plant performance

= 40.1 + 0.00417 Heat Loss (KW) + 0.21460 BOILER Effi (%)
+ 0.8652 Topping plant Effi (%) - 0.119 Bottming plant effi (%)

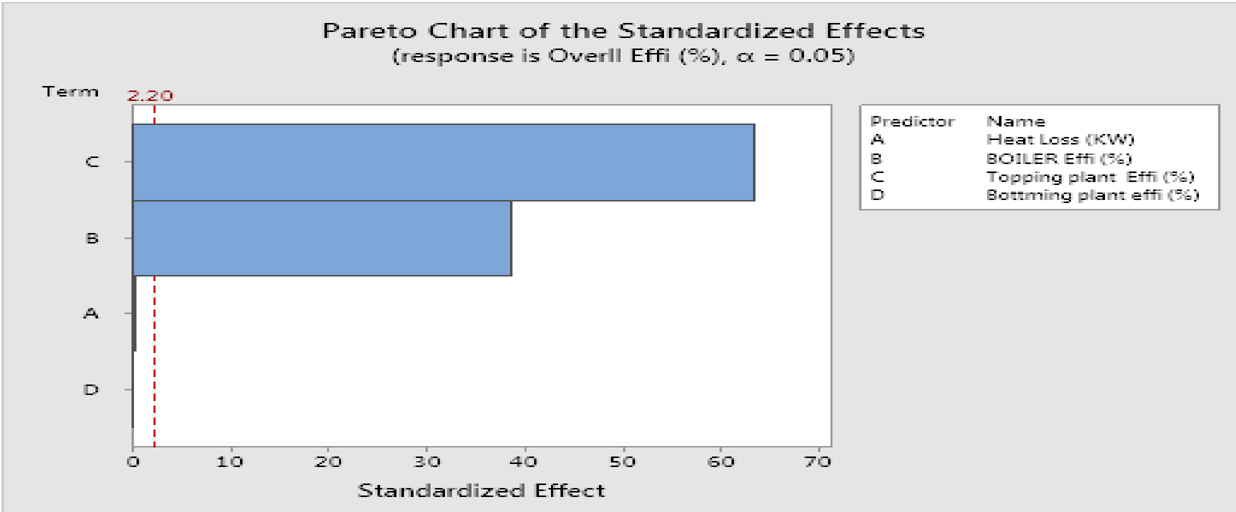

Figure 60: Pareto chart of overall efficiency with heat loss,GT and ST efficiency

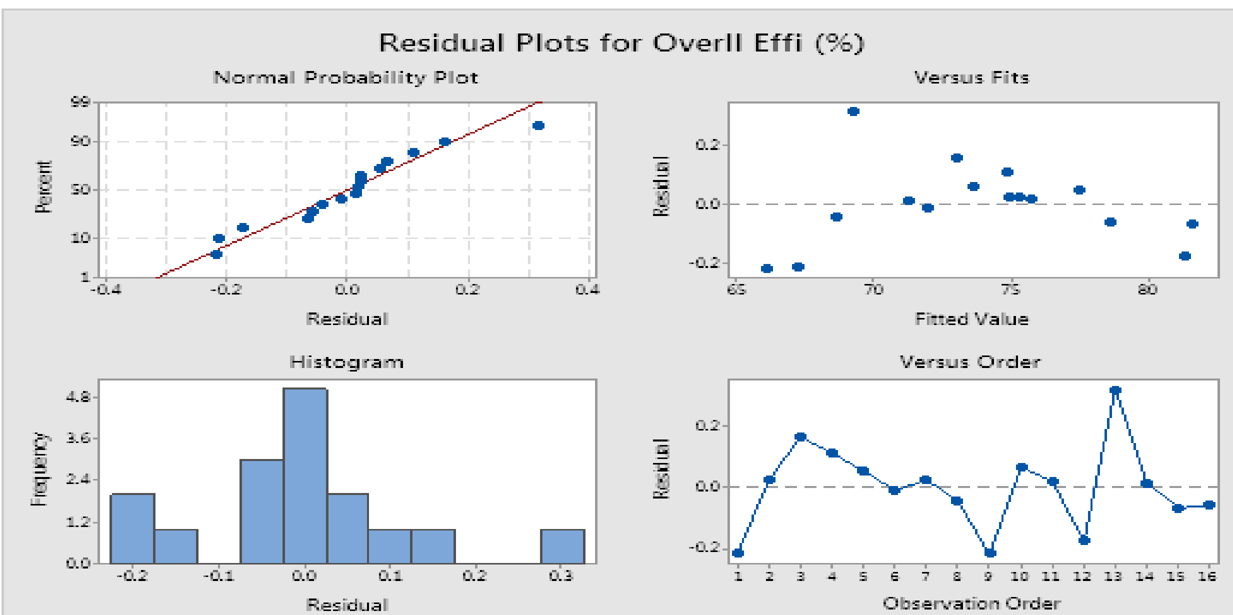

Figure 61: Residual chart of overall efficiency with heat loss,GT and ST efficiency effect

The overall efficiency of plant with the effect of heat loss is shown in fig 60 & 61, the topping plant efficiency is major factor for overall performance, this variation clearly available in fig- 60.

Regression Equation- Overll Effi (%) with effect of GT & ST efficiency

= -172 - 0.2827 Heat Loss (KW) + 1.030 Topping plant Effi (%) + 6.93 Bottming plant effi (%)

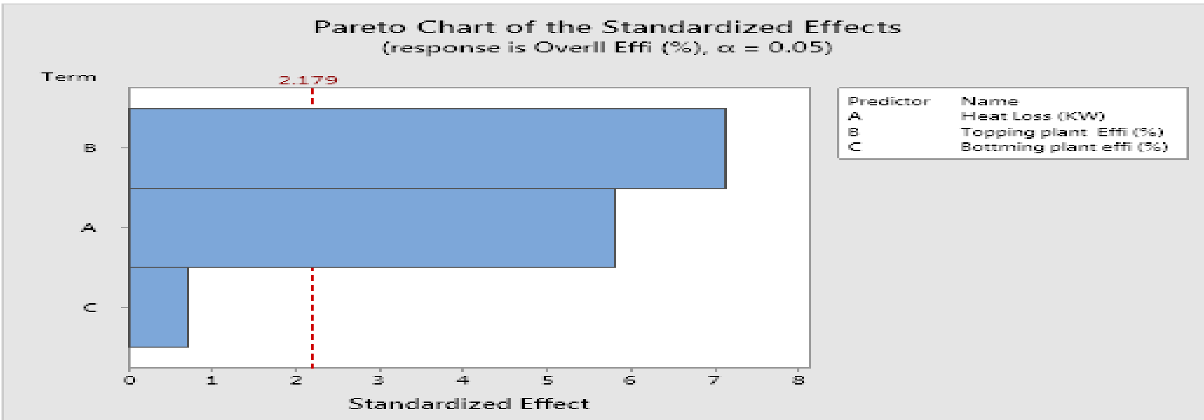

Figure 62: Pareto chart of overall efficiency with effect of GT and ST efficiency effect

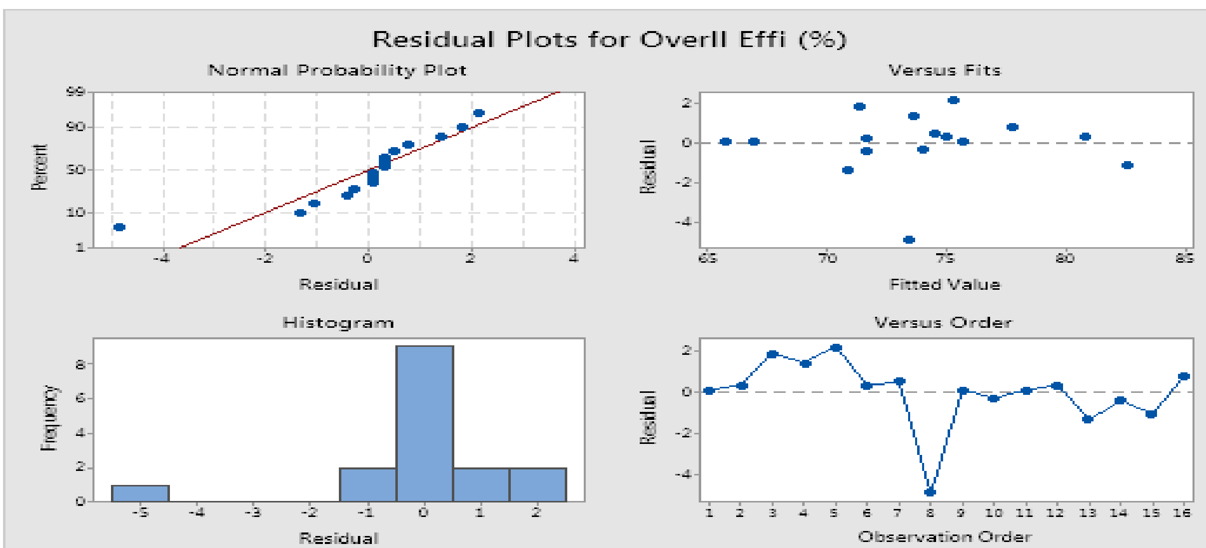

Figure 63: Residual chart of overall efficiency with effect of GT and ST efficiency effect

The GT plant efficiency is significant factor for overall performance, as shown in fig-62, The all test run numbers are statble at nearer to the residual straight line as available in fig-63.

Regression Equation- Overll Effi (%) with effect of boiler efficiency, GT& ST plant efficiency

$$= 38.2 + 0.8668 \text{ Topping plant Effi (\%)} - 0.055 \text{ Bottming plant effi (\%)} + 0.21228 \text{ Boiler Effi (\%)}$$

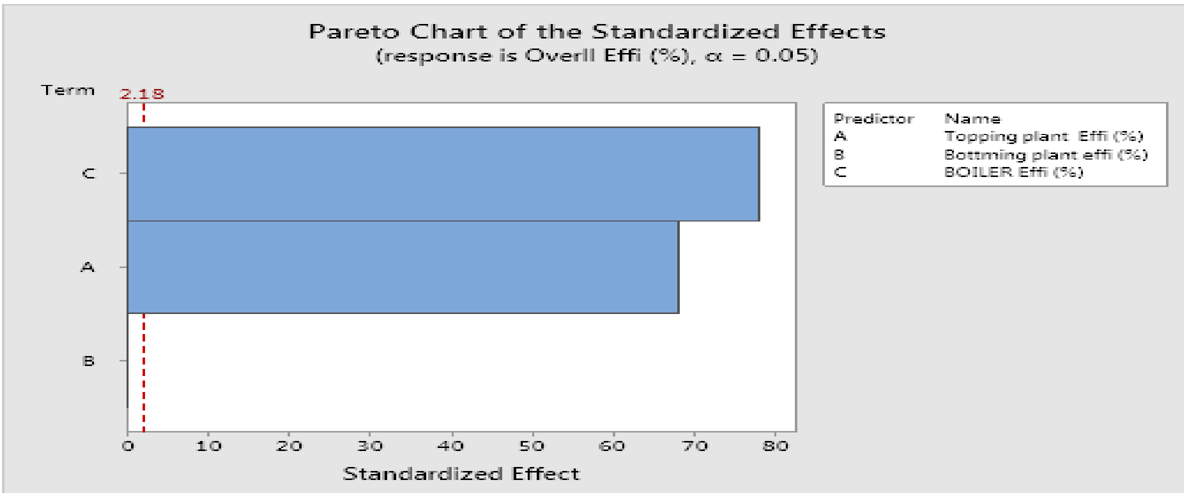

Figure 64: Residual chart of overall efficiency with effect of Boiler efficiency, GT and ST efficiency effect

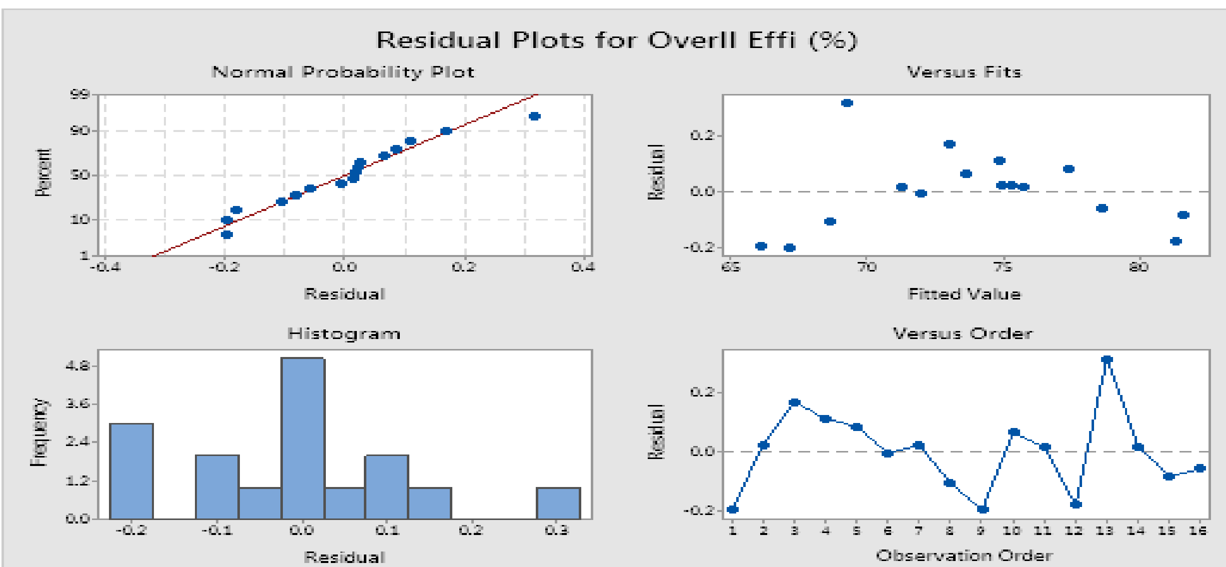

Figure 65: Residual chart of overall efficiency with effect of Boiler efficiency, GT and ST efficiency effect

Fig-64-65shows the effect of boiler efficiency on overall efficiency of combined GT-ST plant. The boiler efficiency has significant effect, and all test run are in residual straight line, so no large deviation on its values.

Regression Equation- Overll Effi (%) with effect of cyclepressure, GTIT and mass of flue gas

$$= 31.79 + 1.161 \text{ Pressure(Bar)} + 0.01690 \text{ Temperature (Degree Celsious)} + 11.80 \text{ CP\_flue gas} - 0.000301 \text{ m\_flue gas}$$

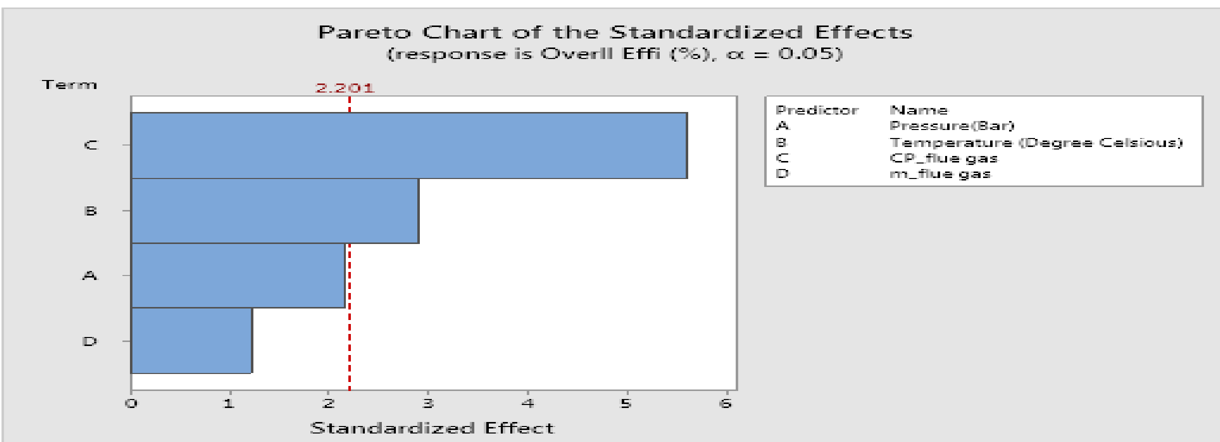

Figure 66: Pareto chart of overall efficiency with effect of cyclic pressure, GTIT and mass of flue gas

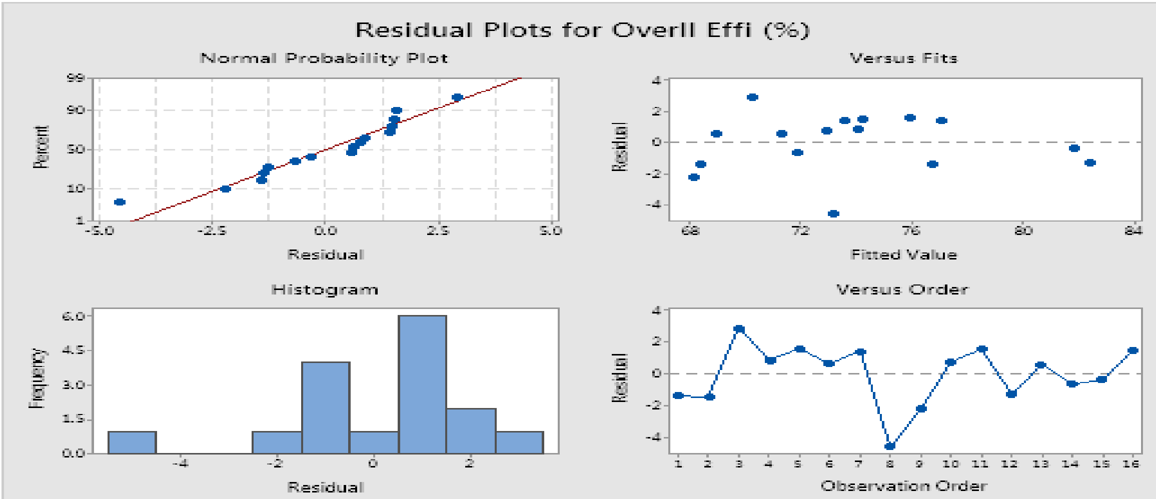

Figure 67: Residual chart of overall efficiency with effect of cyclic pressure, GTIT and mass of flue gas

The effect of cyclic pressure and flow rate of flue gas on GT plant have shown in fig-66, the maximum pressure give the more GTIT and its gives the higher thermal efficiency, the fig-67 provide the residual plot of above mentioned regression equation relation.

Regression Equation- BOILER Effi (%)

BOILER Effi (%) = 275 + 13.1 Pressure(Bar) - 0.444 Temperature (Degree Celsious)
+ 8.00 CP_flue gas - 0.00159 m_flue gas + 0.555 Temp Diff
- 2.04 Heat Loss (KW)

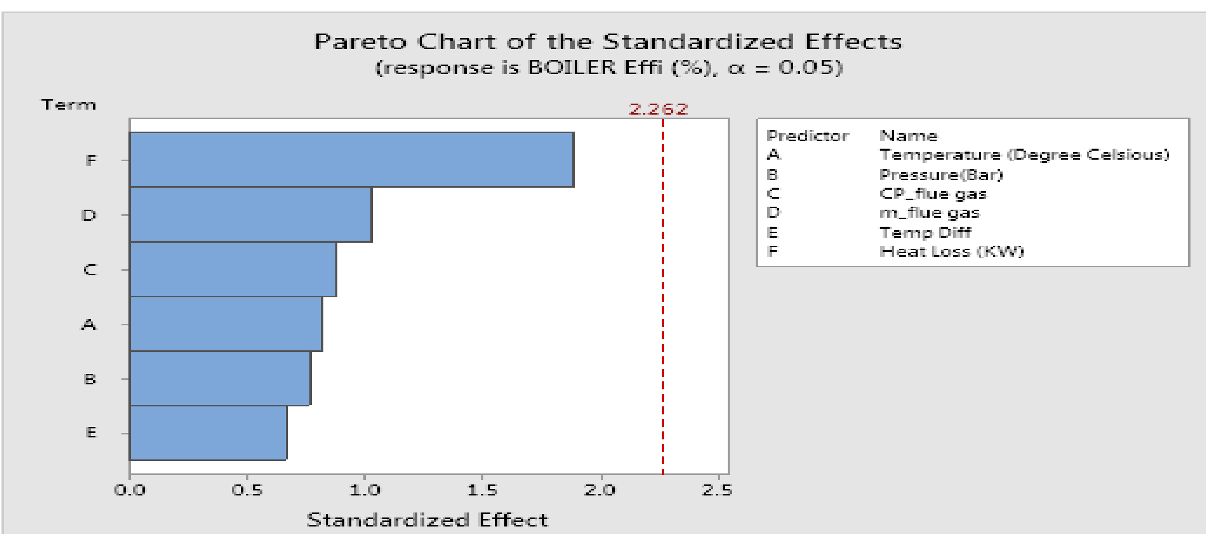

Figure 68: Pareto chart of boiler efficiency

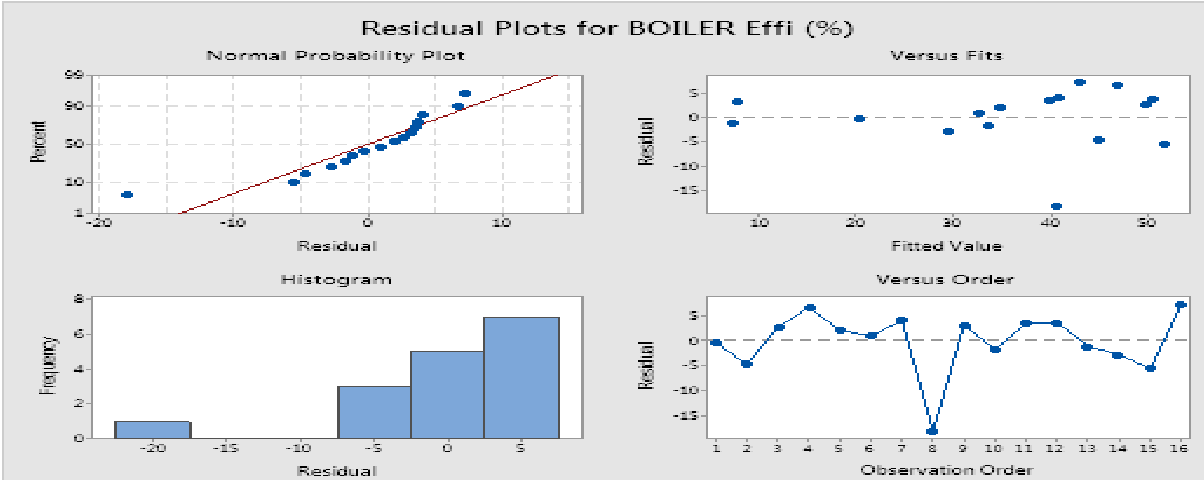

Figure 69: Residual chart of boiler efficiency

The heat loss is influencing factor for boiler performance, more heat loss between GT & ST plant, ST plant efficiency will decreses and further overall performance also will be affected.The fig 68-69 shwing the significant factores for bpiler efficiency as per regression expression. L16 Regression model for Actual and Predicted result comparison is shown in table 12.

Table 12: L16 Regression model for Actual and Predicted result comparison

Test Run	Compressor Pressure (Bar)	Temperature –GTIT (Degree Celsius)	Natural Gas	Actual Overall Efficiency (%)	Predicted Overall Efficiency (%) By DOE	Residual	% Deviation

P1	6	600	Butane	37.2	40.03	-2.83	-7.607526882
P2	6	700	Methane	44.6	43.012	1.588	3.560538117
P3	6	800	Ethane	44.4	45.994	-1.594	-3.59009009
P4	6	900	Octane	45.3	46.976	-1.676	-3.699779249
P5	7	600	Methane	43.07	40.458	2.612	6.064546088
P6	7	700	Butane	41.8	43.44	-1.64	-3.923444976
P7	7	800	Octane	45	46.422	-1.422	-3.16
P8	7	900	Ethane	45.5	49.404	-3.904	-8.58021978
P9	8	600	Ethane	35.5	40.886	-5.386	-15.17183099

P10	8	700	Octane	42.9	43.886	-0.986	-2.298368298
P11	8	800	Butane	45.39	46.85	-1.46	-3.216567526
P12	8	900	Methane	50.5	49.482	1.018	2.015841584
P13	9	600	Octane	38.8	41.314	-2.514	-6.479381443
P14	9	700	Ethane	40.78	44.296	-3.516	-8.621873467
P15	9	800	Methane	49.8	47.278	2.522	5.064257028
P16	9	900	Butane	48.19	50.26	-2.07	-4.295496991

The all performance parameters of combined GT-ST power generation and VARS cooling systems are available in table 11. The boiler efficiency and heat loss between GT & ST plant have been consider for overall output and plant performance. The maximum overall efficiency of plant is achieved when boiler efficiency factor is considered. The heat loss factor signifies the maximum possible energy degradation during the coupling of cycle. The actual efficiency of combined plant is estimated by exergetic performance as 38.5%

which is valuable output for plant operation. The maximum energy losses have been counted in CC of GT plant and exhaust flow unit of ST plant. The performance of VARS cooling system is evaluated by 0.708 which is remarkable performance and able to produce sufficient cooling effect for meeting hall cooling purpose. The additional cooling load in terms of 05 TR or 17.5 KW enhances the overall output and efficiency also. All operational and functional results have listed in table no 12 &13.There is no more deviations between actual results of plant and predicted results from the DOE analysis. Fig- 70 explains the actual and predicted results variation of overall efficiency of combined GT-ST plant.

Table 13: MLR test run and overall performance

Plant Operating Factors	**Heat Rejection and Heat Loss**		**Plant Efficiencies**		**Overall efficiency with boiler efficiency consideration**		**Overall efficiency with heat loss consideration**			**Overall output with heat loss consideration**		**Overall efficiency with heat loss consideration**	
Test run	**Heat rejection from gt plant (KW)**	**Heat loss between gt& st plant (KW)**	**Boiler effi (%)**	**Gt plant efficiency (%)**	**Overall efficiency without cooling(%)**	**Overall efficiency with cooling(%)**	**Overall efficiency with cooling(%)**	**Overall efficiency with cooling(%)**	**Cooling output (MW)**	**Overall output without cooling(MW)**	**Overall output with cooling(MW)**	**Overall output without cooling(MW)**	**Overall efficiency with cooling(MW)**
P1	**64.59**	**51.6**	**20.12**	**30.5**	**66.97**	**67**	**36.2**	**36.22**	**0.0175**	**36.22**	**36.24**	**14.63**	**14.64**
P2	**65.97**	**39.4**	**40.27**	**34.9**	**75.29**	**75.4**	**43.2**	**43.12**	**0.0175**	**43.12**	**43.14**	**6.07**	**6.08**
P3	**65.38**	**31.09**	**52.45**	**29.3**	**73.16**	**73.3**	**42.1**	**41.97**	**0.0175**	**41.97**	**41.99**	**5.03**	**5.047**
P4	**65.15**	**30.3**	**53.49**	**31.2**	**74.97**	**75.2**	**43.7**	**43.52**	**0.0175**	**43.52**	**43.53**	**2.89**	**2.90**
P5	**77.87**	**49.1**	**36.95**	**38.2**	**77.50**	**77.5**	**49.1**	**42.26**	**0.0175**	**42.26**	**42.28**	**28.93**	**28.94**
P6	**65.30**	**43.29**	**33.70**	**32.7**	**71.95**	**72**	**40.4**	**40.38**	**0.0175**	**40.38**	**40.40**	**8.24**	**8.25**

P7	**65.52**	**36.03**	**45.01**	**33.39**	**74.98**	**74.9**	**43.3**	**43.17**	**0.0175**	**43.17**	**43.19**	**4.56**	**4.57**
P8	**40.98**	**31.8**	**22.40**	**31.67**	**68.56**	**68.9**	**43.6**	**43.35**	**0.0175**	**43.35**	**43.37**	**2.65**	**2.66**
P9	**65.04**	**57.95**	**10.91**	**31.5**	**65.89**	**65.9**	**35.3**	**34.87**	**0.0175**	**34.87**	**34.89**	**7**	**7.01**
P10	**64.89**	**44.17**	**31.93**	**35.07**	**73.70**	**73.3**	**41.5**	**41.71**	**0.0175**	**41.71**	**41.73**	**9.3**	**9.31**
P11	**65.46**	**37.1**	**43.32**	**34.7**	**75.75**	**75.8**	**43.8**	**43.72**	**0.0175**	**43.72**	**43.74**	**6.01**	**6.02**
P12	**65.53**	**30**	**54.22**	**38.5**	**81.17**	**80.3**	**49**	**48.58**	**0.0175**	**48.58**	**48.59**	**3.9**	**3.91**
P13	**60.68**	**56.9**	**6.23**	**36.32**	**69.59**	**69.7**	**38.5**	**38.48**	**0.0175**	**38.48**	**38.50**	**20.21**	**20.22**
P14	**64.72**	**47.47**	**26.65**	**33.58**	**71.24**	**71.3**	**38.4**	**39.64**	**0.0175**	**39.64**	**39.66**	**14.26**	**14.27**
P15	**65.53**	**35.26**	**46.19**	**40.8**	**81.55**	**81.4**	**48.6**	**48.46**	**0.0175**	**48.46**	**48.47**	**6.45**	**6.46**
P16	**65.18**	**32.5**	**50.14**	**36.35**	**78.56**	**78.4**	**46.4**	**46.31**	**0.0175**	**46.31**	**46.33**	**4.79**	**4.80**

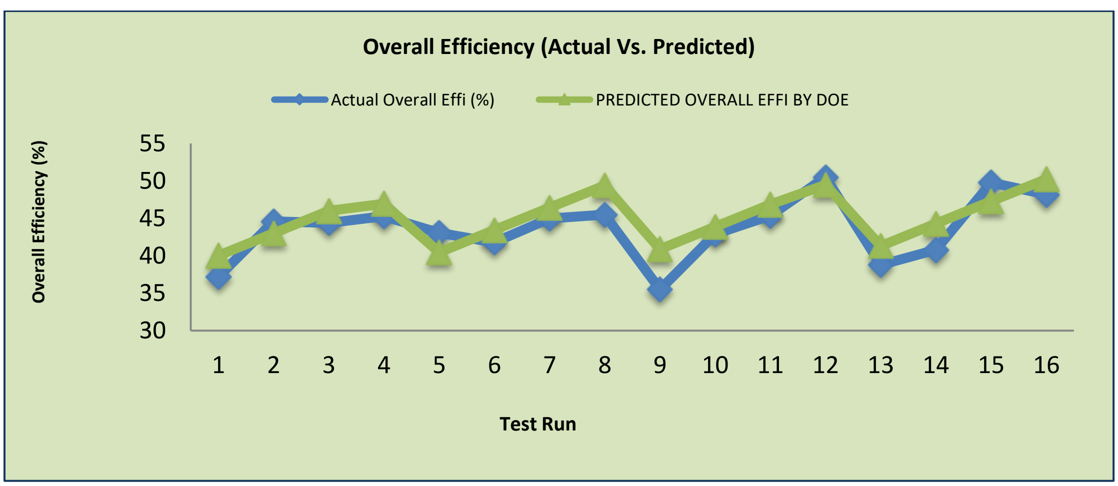

Figure 70: Overall efficiency (Actual Vs Predicted by DOE)

5.3 Result analysis of regeneration-reheating thermal power plant by using steam ejector cooling system

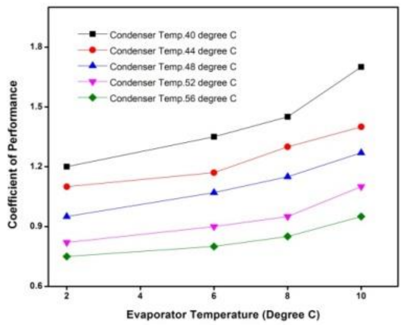

Figure 71: Effect of entrainment ratio on condenser temperature

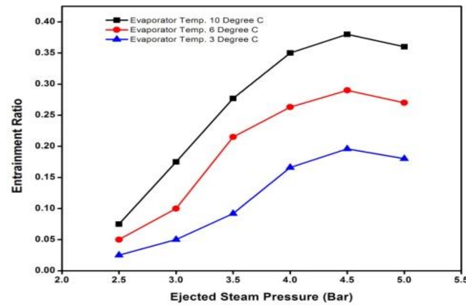

Figure 72: Effect of entrainment ratio on evaporator temperature

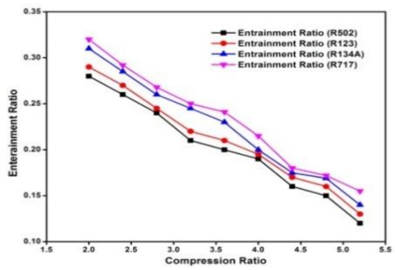

Figure 73: Effect of compression ratio on entrainment ratio

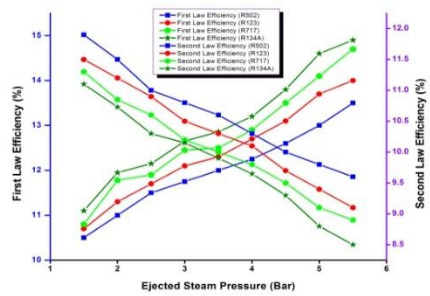

Figure 74: Efficiencies variation with ejected steam pressure

The Effect of entrainment ratio on condenser temperature and evaporator temperature is shown in fig. 71 and 72 respectively. The ammonia and R-134a have performed higher entrainment at low thermo-compression process. Higher compression is responsible for low entrainment. Fig. 73 clearly indicates the effect of entrainment of different eco-friendly refrigerants at compression ratio. The comparative result of 1st law and 2nd law efficiency is available in fig-74, the R134a and ammonia performed better.

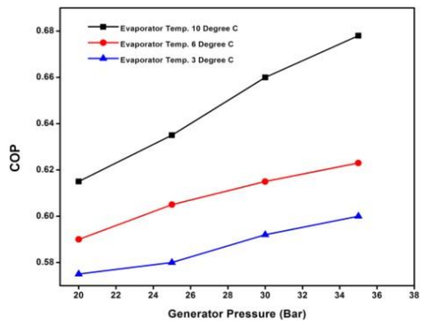

Figure 75: Effect of generator pressure on COP at different absorber temp

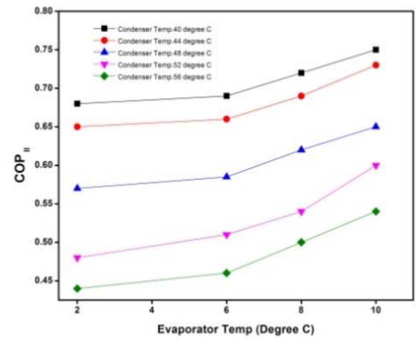

Figure 76: Effect of evaporator temperature on COP at different condenser temp

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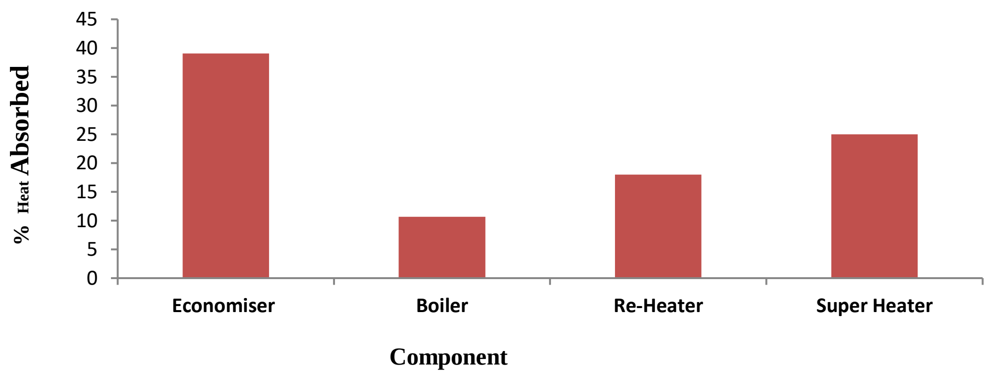


Figure 77: Components Heat Absorbed of Combined Reheating Regenerative Rankine cycle

The Effect of generator pressure on COP at different absorber temp is given in Fig.75 and Effect of evaporator temperature on COP at different condenser temp is shown in fig. 76. Also, the heat absorbed by the Components of Combined Reheating Regenerative Rankine cycle is shown in fig.77.

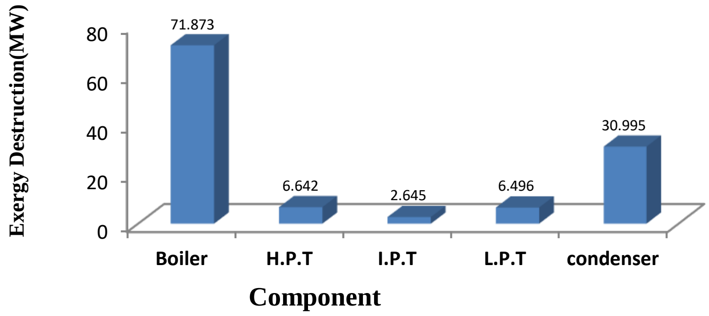


Figure 78: Exergy Destruction in components of Combined Reheating Regenerative Rankine cycle

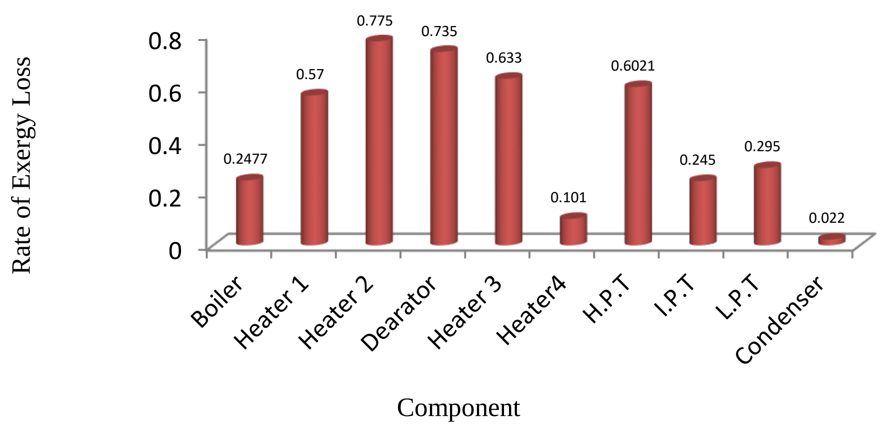


Figure 79: Exergy Loss in components of Combined Reheating Regenerative Rankine cycle

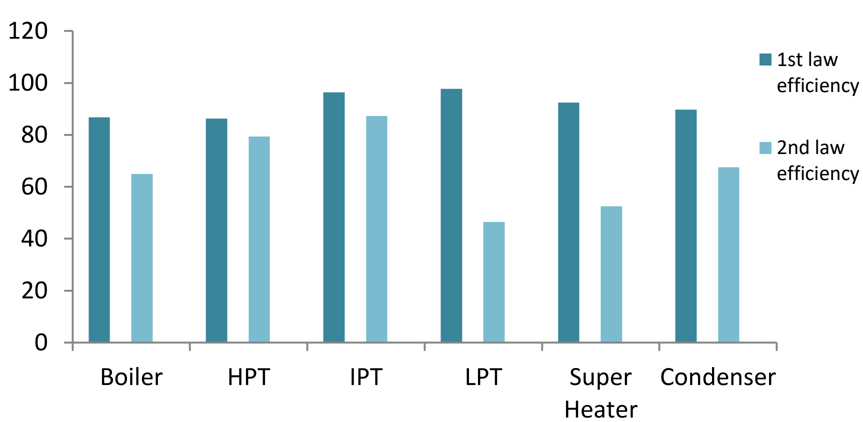


Figure 80: Exergy Loss in components of Combined Reheating Regenerative Rankine cycle

Exergy destruction of boiler is maximum with respect to condenser and turbine respectively which is clearly shown in figure 78. The rate of exergy losses have been estimated of different components of plant.The 60% to 77% of exergy losses founded in high pressure turbine ,Heater 12,3 and deaerator which is shown in figure 79. On comparing figures 79, the Boiler, turbine (HPT), heaters and de-aerator are to be redesigned due to major exergy loss.

The efficiencies comparison between plant components is clearly shown in figure-80, the low pressure turbine has considerable gap between 1st and 2nd law efficiencies. So there is an opportunity of improving low pressure turbine efficiency by decreasing irreversibility. Table 14 and 15 explains the efficiency observation and exergy results of major unit of plant.

Table 14: Component Efficiencies (η1 & ηII) comparison of combined Reheating-Regenerative Rankine Thermal Power Plant

Component	**First law efficiency (η_1)**	**Second law efficiency (η_{II})**
Boiler	86.80%	64.90%
High pressure turbine (HPT)	86.25%	79.39%
Intermediate pressure turbine (IPT)	96.40%	87.20%
Low pressure turbine (LPT)	97.67%	46.45%
Super heater	92.39%	52.50%
Condenser	89.77%	67.47%

Table 15: Exergy observation of components of combined reheating-regenerative rankine power system

Components	**Exergy ($\in$)output/**	$\Delta s = c_p \ln(\Delta T_{out} \vert \Delta T_{in}$	**IRR**	**Rate of exergy**	**Rate of**	**EDD ratio =**

	input $= \mathbf{m} \times \mathbf{c_p} \times \mathbf{T_o}\left[\frac{\mathbf{T_b}}{\mathbf{T_o}} - \mathbf{1} - \mathbf{ln}\frac{\mathbf{T_b}}{\mathbf{T_o}}\right]$			$= \mathbf{To\sum \Delta s}$	**decrease =** $\in_{\mathbf{output}} - \in_{\mathbf{input}}$	**exergy loss** $= \frac{\in_{\mathbf{in}}}{\in_{\mathbf{out}}}$	**IRR/Qin (Qin means Qboiler)**
	$\in_{\mathrm{output}}$ (MW)	$\in_{\mathrm{in}}$ (MW)	(kj/kgk)	(kj/kg)	MW		
Boiler	25.13	21.28	0.125	37.25	3.85	0.846	0.014
Reheater	30.4	15.66	0.487	145.3	14.74	0.515	0.054
Super Heater	49	25.13	0.517	154.3	23.97	0.512	0.057
Economiser	21.28	12.07	0.327	97.7	9.21	0.571	0.036
HPT	29.861	49.589	0.411	122.47	19.72	0.60	0.045
IPT	7.622	30.986	0.874	260.45	23.32	0.246	0.097
LPT	2.557	8.666	0.557	165.98	6.11	0.290	0.061
Condenser	0.048	2.157	0.6477	193.0146	2.109	44.6	08.98

Heater1	1.9	1.083	0.606	180.5	0.817	0.98	.006
Heater2	0.004	0.003 11	0.315	93.8	0.0009	0.77	0.034
Heater 3	0.071	0.045	0.250	74.5	.026	0.63	0.027
Heater 4	0.076	0.007 7	0.559	166.58	0.068	0.10	0.058
Dearator	1.105	0.813	0.207	61.66	0.292	0.73	0.022

5.4 Result analysis solar thermal based vapour adsorption system for condenser heat recovery

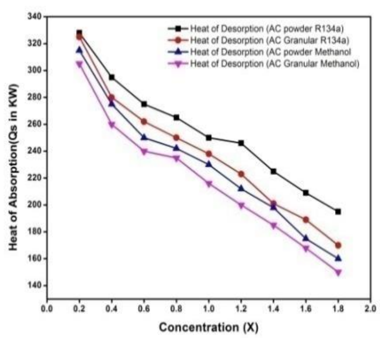


Figure 81: Effect of concentration and heat of absorption

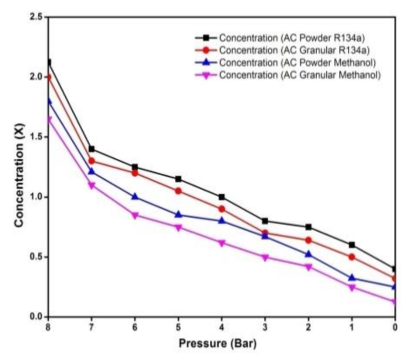
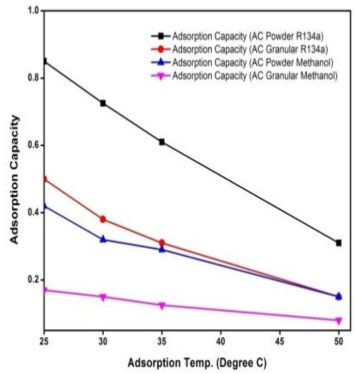


Figure 82: Effect of pressure inside adsorber generator and concentration



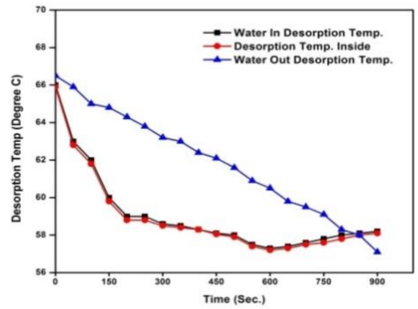


Figure 83: Effect of adsorption temperature and adsorption capacity

Figure 84: Effect of adsorption time and desorption temperature

Fig. 81 illustrates the relation between the adsorption capacity .of the all tested pairs. As shown in the figure, the adsorption capacity decreased with increase the adsorption temperature.

Fig. 82&83 presents the relation between the measured temperatures inside the adsorber, the atmosphere temperature and water outlet from the adsorber. Adsorption temperature is set to 25°C. From the figure,. This is due to condensation of refrigerant molecules inside the adsorbent pores. As the time pass the temperature is decreased, and, this due to there is no more refrigerant

adsorbed. Also, from the figure 84 the water outlet temperature is increased with the time and this due to heat transferred from the adsorbent as it’s the heat of adsorption, this increase in water outlet temperature continued even with the decrease in the temperature inside the adsorber, this is because the temperature inside the adsorber is still higher than water outlet temperature.

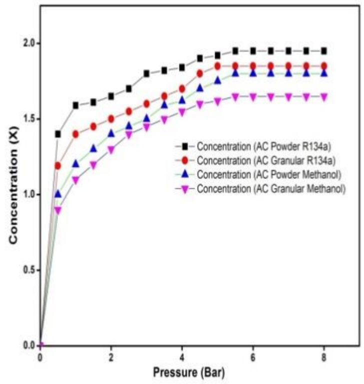


Figure 85: Effect of pressure on concentration

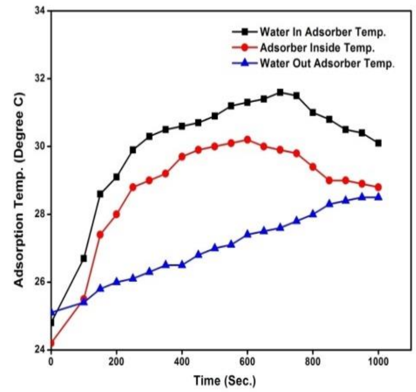


Figure 86: Effect of adsorption time on adsorption temperature

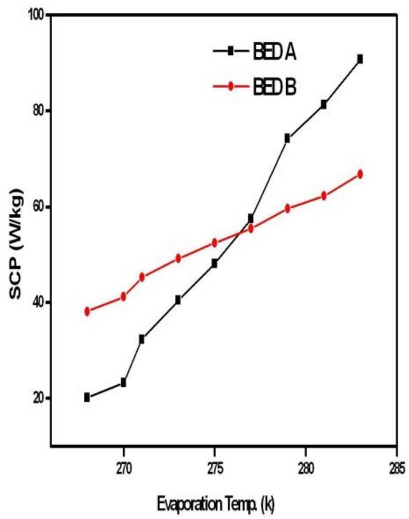


Figure 87: Effect of pressure on concentration

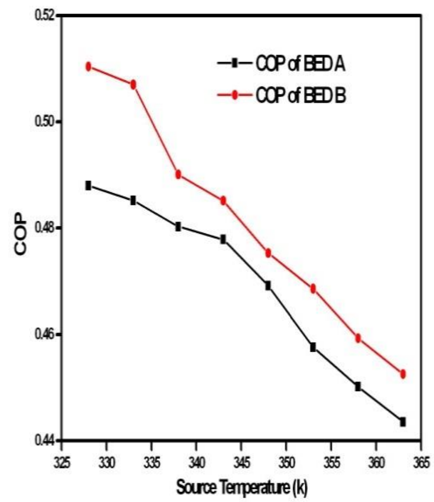
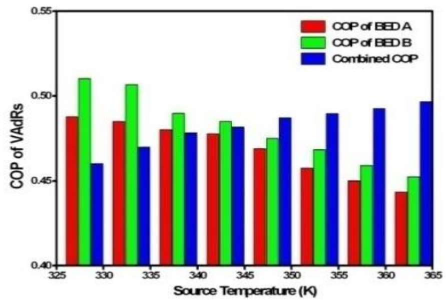


Figure 88: Effect of adsorption time on adsorption temperature



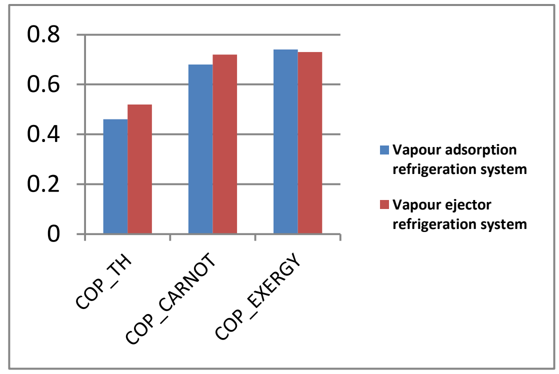


Figure 89: Effect of source temperature on COP

Figure 90: Comparative COP of vapor adsorption and ejector cooling system

Fig.85&86 explores the variation of pressure inside the adsorption chamber and concentration of adsorbent-adsorbate. Saha and Akshay model [18, 19] reported same variation with the different granular size of activated-carbon. It causes the heat transfer during adsorption and desorption process inside the chamber.

It is found that when the adsorption temperature is low the heat of adsorption is high for both pair of adsorbent-adsorbate. The capacity of adsorption is high at low adsorption temperature. The effect of heat of adsorption and temperature is shown in fig. 87&88. Higher generator temperature produces higher pressure and superheated vapour of refrigerant, performance of refrigeration is function of generator temperature and pressure, but more temperature also causes of heat loss and decreases the COP also. The effect of source temperature on COP is given by fig.89.

The COP of ejector refrigeration is slightly more than vapour adsorption for R-134a refrigerant with same 10TR of cooling effect. Table 16 and 17 shows theperformance of double bed VADRS system and table 18 concludes all parametric output of double bed VADRS system. Fig. 90 explains COP of both refrigeration systems.

Table 16: Performance of Adsorption bed-1

COP_{TH}	**Q_{COOL}(kJ)**	**Q_{HEAT}(kJ)**	**Q_{REF}(kJ)**	**SCP(kW/kg)**	**T_{GEN} (K)**
0.488	854.6	33543	17222	23.32	328
0.4852	1688	58643	30139	40.53	333
0.4803	2687	84050	43056	57.51	338
0.4749	3850	109765	55973	74.25	343
0.4692	5179	135787	68890	90.76	348

Table 17: Performance of Adsorption bed-2

COP_{SOLAR}	**COP_{TH}**	**I_G (W/m^2)**	**Q_{COOL}(kJ)**	**Q_{HEAT}(kJ)**	**Q_{REF} (kJ)**	**SCP (kW/kg)**	**T_{GEN}(K)**
0.4811	0.507	750	1570	19858	11638	38.18	333
0.5529	0.482	775	1865	24804	13820	45.34	340.5
0.6202	0.4593	800	2159	30136	16002	52.49	348
0.6834	0.4387	825	2454	35855	18184	59.65	355.5
0.7429	0.4199	850	2748	41961	20366	66.81	363

Table 18: Parametric Results of combined System

Performance Parameter	**Resultant Values**

Turbine work output (W_{HPT}& W_{LPT})	294.7 & 1072 kJ/kg
Pump Work (W_{PUMP})	16.23 kJ/kg
Boiler Heat Generation (Q_{IN})	3484 kJ/kg
Condenser heat exhaust (Q_{OUT})	2133 kJ/kg
Plant Efficiency (η_P)	0.388
Net Work Output (W_{NET})	1350 kJ/kg
COP of combined VAdRS ($COP_{COMBINEDVADRS}$)	0.46 to 0.51 (13-15% increment in overall COP)
Sp.Cooling Power ($SCP_{COMBINEDVADRS}$)	70-90 W/kg (10-16% increment in overall SCP)
COP gain by Solar energy (COP_{SOLAR})	0.48-0.74 (most effective COP)

CHAPTER 6

Chapter-6 Conclusion and future scope

6.1 Conclusions

The detailed thermodynamic (energy & exergy) analysis of four different heat recovery systems have been studied. The theoretical investigation of live plant data of power generation, milk plant has done by using ORC,VARS Vapour adsorption and ejector refrigeration systems for combined cooling, heating and power generation effect. The complete energy and exergy analysis are carried out for all proposed systems. The statistical analysis also performed for error estimation, parametric comparison and optimization for both actual and model data. The outcomes of the present research work cover majorly followings-

A- Energy - exergy analysis of thermal systems.

B-Performance of different thermodynamic system for combine cooling-heating and power generation.

C-Statistical analysis-based optimization methods for parametric comparison.

D-Solar thermal integrated waste heat recovery systems.

The summery of present research work have concluded below

1-The quantitative energy is estimated by 1st law of thermodynamics and the actual useful energy is estimated by the 2nd law of thermodynamics. The exergy analysis give the real performance of thermal system and able to identify the individual component loss, component performance and further treatment of process, scale up the process where losses are more. The maximum exergy loss has been found 66% in CC of GT plant where as exhaust flue gas unit has considerable exergy loss as 13.5% due to massive amount of heat is generated and discharged also.The steam turbine has higher energy destruction as compare to gas-turbine unit due to some amount of steam get condensed during steam expansion in steam turbine, and heat lost during the phase change from steam to water moisture, there is no any phase change is possible in gas turbine expansion. The 70% of exergy loss also found in feed water heater (De-aerator, FWH) of combined reheating-regeneration steam power generation plant. The solar thermal energy integrated component of heat recovery has considerable irreversibility during the hot water generation, the 7-8% of energy loss is found when solar system is operate for generator of VAM and adsorption cooling systems. Exergy loss or irreversibility greatly affects the system performance. Energy-Exergy estimation helps for components loss identification, design of balance of system as per losses and as well for economic of thermal systems.

2-The solar and without solar operated R134a based ORC for condenser heat recovery of coal fired thermal plant is investigated. The thermal efficiency is achieved 2-6% which is more than without solar operated ORC. The considerable refrigeration effect is found 12 TR. The stack flow heat recovery of combined GT-ST power generation plant is possible by using LibR-H2O based VAM refrigeration and achieves 05 TR of sufficient cooling effect. The combined GT-ST plant exergy analysis have been found the maximum exergy destruction in CC of GT plant about 67%, stack flow unit 13%. The projected LiBr-H2O based VARS is non-toxic, zero consumption of electricity and low value of GWP & ODP factors of environmental safety. This system reduces 162 ton of CO2 yearly when it operates at 08 working hrs, saves 60 units of electricity daily and able to map the decarbonizes economy for commercial infrastructure. The four types of natural gases have been introduced in CC of GT unit at different compression pressure and GTIT operating condition. The maximum possible output and efficiency is achieved by the combustion of methane gas at 8 bar of pressure and 900 0C of GTIT. This operating condition reduces the heat loss between both power cycles and achieves remarkable stack flow temperature of about 262 ^{0}C,which is sufficient for VARS generator temperature and produce cooling effect. The ejector-based cooling system is theoretically investigated for heat recovery of regeneration-reheating rankine plant with different eco-friendly refrigerants. The steam ejector cooling system is also significant solution for heat recovery of captive industry, milk industry. This theoretical analysis considers four eco-friendly refrigerant materials at different operating conditions. The ammonia and R134a performed better than other refrigerants at 5-10 0C of cooling temperature. The steam ejector cooling system has higher performance in terms of COP_carnot and COP_exergy than vapour adsorption refrigeration because desorption process takes some time for adsorption in adsorbent, so it unable to produce continuous cooling with single stage of system. The combination of activated carbon-R134a is suitable pair for heat recovery purpose in vapour adsorption system. The generator temperature is key input for compressor less refrigeration system. R134a gives remarkable COP at higher T_Genarator and pressure. High adsorption capacity of RI34a is also responsible for cooling effect. The COP of ejector refrigeration is slightly more than vapour adsorption for R-134a refrigerant with same 10TR of cooling effect.Therefore The activated carbon powder/R-134a pair is highly recommended to be used as adsorption refrigeration working pair because of its higher maximum adsorption capacity than the other tested pairs, to produce a compact, efficient and reliable for long life performance adsorption refrigeration system.

3-The implementation of DOE approach helps for parametric optimization and comparison between tested & predicted value of GT plant. The four different combustible hydrocarbons (methane, ethane, butane and octane) have been tested in combustion chamber at different pressure ratio (6,7,8&9 bar) and operating temperature ranges (600,700,800 & 900 ^{0}C) of GTIT. MLR model helps to identify the suitable combination of input factors for optimized result of plant operation. The best possible performance is achieved by the combination of 09 bar of compression ratio with 900 ^{0}C of gas turbine inlet temperature of maximum temperature of GT-ST plant gives the 47.6% of overall efficiency by using methane gas as a fuel in CC of gas turbine and at same combination estimate 30.6% of minimum heat loss. The MLR statistical highlight one key fact, that is, the linear model is statistically acceptable due to the best-fit line when plotted on a data set of 16 test runs. This indicates that the predicted values and the real data are in close proximity to each other. The variation found in actual results from the predicted in terms of average absolute residuals and average value of % deviation values are -1.328 and -3.371 respectively for the case of overall efficiency analysis of plant. It was due to cyclic variations that accounted for this minor difference between the real and the predicted data.

4- The employment of solar energy in terms of thermal energy field provide remarkable energy source as generator heat source for ORC, vapour adsorption refrigeration and vapour absorption refrigeration systems. The heat input by solar thermal field enhances the generator temperature and it helps to superheating of refrigerants for cooling effect. The integrated solar thermal system also contributes for renewable energy source based intelligent heating-cooling and power generation technology.

6.2 Recommendation for future

The thermodynamic (energy-exergy) analysis of all presented combined heating-cooling and power generation systems are theoretically investigated with live data of few Indian power generation plants. These investigations have two-fold benefits for the researchers and private sector developers in the way that the implementation of alternative refrigeration systems for cooling effect in efficient manner. The opportunities of thermal system for unused heat recovery with the integration of solar thermal systems give the valuable output in terms of process heating, expander work (power output), refrigeration effect etc. The all mentioned thermal system does not consume high grade energy. It operates with waste heat of plant or solar thermal energy source; no toxic emission takes place and is able to map the decarbonize economy with control of carbon discharge. The combine energy-exergy and DOE analysis of proposed thermodynamic systems help to identify the effective parameters, system internal losses which contributes to optimize the system. The stack flow heat recovery (above 250 ^{0}C of stack exhaust temperature) and LPT exhaust steam heat (about 160^{0}C) of regenerative thermal power plant can be investigated further in future.

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PUBLICATIONS

PUBLICATION DETAIL

NATIONAL/INTERNATIONAL JOURNAL PUBLICATION

1) Dubey K.K., Mishra R.S. (2020) Comparative Analysis of Combined Gas Turbine–Steam Turbine Power Cycle Performance by Using Entropy Generation and Statistical Methodology. Applications of Machine Learning. Algorithms for Intelligent Systems. Springer, Singapore, May 2020, DOI-https://doi.org/10.1007/978-981-15-3357-0_11, Print ISBN978-981-15-3356-3Online ISBN978-981-15-3357-0, PP 157-175.

2) Kaushalendra.Kr Dubey,R.S.Mishra, Energy-Exergy Analysis of Combined Reheating-Regenerative Rankine Cycle using Entropy Generation Principle, International Journal of Recent Technology and Engineering (IJRTE),Volume-8 Issue-3, September 2019 DOI:10.35940/ijrte.C5008.098319, https://www.ijrte.org/download/volume-8- issue-3.(scopus).

3) Kaushalendra.Kr Dubey,R.S.Mishra Statistical Analysis for Parametric optimization of Gas Turbine-Steam Turbine Combined Power Cycle with Different Natural Gas Combustion, International Journal of Recent Technology and Engineering (IJRTE),Volume-8 Issue-3\4, November 2019, DOI:10.35940/ijrte.D8507.118419 https://www.ijrte.org/download/volume-8- issue-4/ (scopus).

4) Kaushalendra.Kr Dubey,R.S.Mishra, Thermodynamic (Energy-Exergy) Analysis of Combined Coal based Thermal Power Plant and Solar Integrated Double Bed Vapor Adsorption System for Heat Recovery and Space Cooling , Saudi Journal of Engineering and Technology, Scholars Middle East Publishers, ISSN 2415-6272 (Print). ISSN 2415-6264 (Online) Pp286-298. Vol-2, Issue-7 July 2017,DOI:10.21276/sjeat (sci-extended).

5) Kaushalendra.Kr Dubey, R.S.Mishra, Solar Integrated Combined Cooling-Power Generation Systems for Waste Heat Recovery Using Different Energy Efficient Materials (extended version of IMAT3E2018 conference), Special Issue of international journal of digital signals and smart systems 2020, Inderscience (accepted and process of publication) https://www.inderscience.com/info/ingeneral/forthcoming.php?jcode=ijdsss

6) Kaushalendra.Kr Dubey,R.S.Mishra, Industrial And Power Plant Waste Heat Recovery Systems For Combined Cooling-Heating And Power Generation, International Journal of Advance Research and Innovative Ideas in Education,Vol-3,Issue-3, ISSN NO (O):2395-4396,pp-3682-94,june 2017.(UGC, Research Gate) DUI: 16.0415/IJARIIE-5742

7) Kaushalendra.Kr Dubey, R.S.Mishra, Condenser Heat Recovery For Combined Cooling- Heating and Power Generation using Isentropic Fluid, Europian Journal of Engg Research and Science (Belgium),Vol-2,N0-6,e-issn-2506-8016,pp-18-26,june 2017. DOI: http://dx.doi.org/10.24018/ejers.2017.2.6.360

8) Kaushalendra.Kr Dubey,R.S.Mishra, Thermodynamic (energy-exergy) analysis of 9MW Coal based thermal power plant using entropy generation principle, American Journal of/ Engg Research (Stamford),ISSN NO (p):2320-0936. Volume-6, Issue-7, pp52-57,2017.http://www.ajer.org/papers/v6(07)/I06075257.pdf

9) Kaushalendra.K.Dubey,R.S.Mishra. Thermodynamic (Energy-Exergy) analysis of solar integrated power-cooling combined generation system .International Journal of Advance Research and Innovation, ISSN 2347 – 3258.Volume 4,Issue 31(2016),129-135

10) Kaushalendra.Kr Dubey, R.S.Mishra "Energy analysis of combined gas turbine-steam turbine power Generation with integrated sustainable cooling system for plant Stack flow heat recovery and carbon economy"Elsevier, Materials Today.EMSME2020, NIT Delhi. (accepted).

NATIONAL/INTERNATIONAL CONFERENCE PUBLICATION

1 Kaushalendra Kumar Dubey, R.S Mishra ,Solar integrated energy efficient systems for waste heat recovery of power generation industry using different eco-friendly materials, , International meeting on advanced technology in energy and electrical engg on Clean Energy, (IMAT3E-2018), Fez,Morococo,22-23 November 2018. https://doi.org/10.5339/qproc.2019.imat3e2018.1 Available in online https://hal.archives-ouverts.fr/hal-01936981). (Shortlisted in inderscience journal of IJSSDS).

2 Kaushalendra Kumar Dubey, R.S Mishra , A Review on Properties of Phase Change Material for Solar Thermal Storage System, International Conference on advanced research and innovation (ICARI), Organized by International Journal of advanced research and innovation at IEI Bhawan, New Delhi,01 February, 2014.

3,R.S Mishra, Kaushalendra Kumar Dubey , Thermodynamic (Energy-Exergy) Analysis of Solar Assisted Power-Cooling Combined Generation SystemsInternational Conference on advanced research and innovation (ICARI), Organized by International Journal of advanced research and innovation at IEIBhawan, New Delhi. 27 February 2016..

4 R.S Mishra, Kaushalendra Kumar Dubey Thermodynamic Study of Combined Cooling Heating and Power Generation by Condenser Heat Recovery Using R-134a International Conference on advanced research and innovation (ICARI), Organized by International Journal of advanced research and innovation at IEI Bhawan, New Delhi,20 January, 2019.

5 R.S Mishra, Kaushalendra Kumar Dubey, Energy Efficient Thermodynamic Systems For Modern Power Generation Infrastructure And Decarbonized Economy,International Conference On “Innovative Research in Applied Physical, Chemical, Mathematical Sciences, Statistics and Emerging Energy Technology for Sustainable Development” (APCMSET-2020) Organized by “Krishi Sanskriti Publications”JNU Delhi,14 june 2020.(paper submitted).

SUMMERY OF PUBLICATION

Journal Publications				
S.No	**Title Of Paper**	**Name Of The Authors**	**Name Of Journal**	**Indexing status of journal with indexing agency**
1	Comperative Analysis Of Combined Gas Turbine-Steam Turbine Power Cycle Perfromance By Using Entropy Generation And Statistical Methodology	Kaushalendra Kumar Dubey, R.S.Mishra	**Chapter Of Book Titled "Elements Of Statistical Learning" In Springer Nature (Singapore) Book Series "**Artificial Machine Learning-ISBN No-978-981-15-3356-3)**" (Published-**https://link.springer.com/chapter/10.1007/978-981-15-3357-0_11**)**	SCI
2	Energy-Exergy Analysis Of Combined Reheating-Regenerative Rankine Cycle Using Entropy Generation Principle	Kaushalendra Kumar Dubey, R.S.Mishra	International Journal Of Recent Technology And Engineering (IJRTE), Sept. 2019,DOI:10.35940/ijrte.C5008.098319,https://www.ijrte.org/download/volume-8-issue-3	SCOPUS
3	Statistical Analysis for Parametric optimization of Gas Turbine-Steam Turbine Combined Power Cycle with Different Natural Gas Combustion	Kaushalendra Kumar Dubey, R.S.Mishra	International Journal Of Recent Technology And Engineering (IJRTE) [Nov 2019] DOI:10.35940/ijrte.D8507.118419,https://www.ijrte.org/download/volume-8- issue-4/	SCOPUS
4	Solar Integrated Combined Cooling-Power Generation Systems for Waste Heat Recovery	R.S.Mishra	Special Issue Of IJDSSS, Inderscience Publication. https://www.inderscience.co	Research Gate/S

	Using Different Energy Efficient Materials	kaushalendra Dubey	m/info/ingeneral/forthcoming.php?jcode=ijdsss	COPUS
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