PERFORMANCE ANALYSIS OF COMBINED GAS TURBINE-STEAM TURBINE POWER GENERATION CYCLE

Submitted in partial fulfillment of the requirements
Of the degree of

BACHELOR OF TECHNOLOGY
IN
MECHANICAL ENGINEERING

By

KHURSHED ALAM (1614101084)
MANISH KUMAR GUPTA (1614101373)
MD SHAMIM ANSARI (1614101097)
SAIF UL ISLAM (1614101152)

Supervisor:

Mr. K.K dubey

SCHOOL OF MCHANICAL ENGINEERING
GALGOTIAS UNIVERSITY
GREATER NOIDA
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CERTIFICATE

This is to certify that the Research work titled **PERFORMANCE ANALYSIS OF COMBINED GAS TURBINE-STEAM TURBINE POWER GENERATION CYCLE** that is being submitted by **Khurshed Alam, Manish Kr Gupta,Md shamim Ansari and Saif ul Islam** is in partial fulfillment of the requirements for the award of **Bachelor of Technology**, is a record of bonafide work done under my guidance. The contents of this research work, in full or in parts, have neither been taken from any other source nor have been submitted to any other Institute or University for award of any degree or diploma.

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This thesis/dissertation/project report entitled **Performance analysis of combined GT-ST power generation cycle** by **Khurshed alam, Manish Kr Gupta, Md Shamim Ansari and Saif ul Islam** is approved for the degree of bachelor of technology in mechanical engineering.

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The contributions of many different people, in their different ways, have made this possible. I would like to extend my gratitude to the following.

We are grateful to my supervisor ……..

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(Khurshed Alam)

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(Saif ul Islam)

(Department of Mechanical engineering)

ABSTRACT

A combined-cycle power plant basically uses a gas and a steam turbine both together to produce maximum amount of electricity from the same fuel than a traditional simple-cycle plant. The waste heat generated by gas turbine is routed to the nearby steam turbine, which generates extra power. Present GT-ST plant works under a specific hydrocarbon gas like Methane, Ethane, Propane and Butane and perform as per fuel gas available. The proposed title of work focus on the complete thermal (Performance parameter and energy losses) analysis by using different combustion gases and its different operating temperature and pressure condition and analysis of best suitable operating criteria useful gas with the help of DOE (Design of experiment) approach.

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List of abbreviations

1. GT-ST	Gas Turbine-Steam turbine.
2. DOE	Design of experiment.
3. EES	Engineering equation solver.
4. GTIT	Gas turbine inlet temperature.
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Introduction

1.1 Project background

Present GT-ST plant works under a specific hydrocarbon gas like Methane, Ethane, Propane and Butane and perform as per fuel gas available. The proposed title of work focus on the complete thermal (Performance parameter and energy losses) analysis by using different combustion gases and its different operating temperature and pressure condition and analysis of best suitable operating criteria useful gas with the help of DOE (Design of experiment) approach.

1.2 Research purpose and meaning

The main purpose of the research to complete thermal analysis of the system by using different combustion gases and its different operating condition like different temperature and different compression ratio and find the suitable and most efficient fuel who are better in any aspect and analysis of best suitable operating criteria useful gas by using design of experiment software.

1.3 Objective of the study

The main objective of the study to estimate the performance of GT, ST and GT-ST with identify of losses in plant machine by using entropy generation concept by using EES software and developed the energy exergy model of combined GT-ST plant with actual performance analysis.

2

Literature review

2.1 Introduction

The term “combined cycle” refers to the combination of multiple thermodynamic cycle to generate electricity or power. Gas turbine produce work using Brayton cycle and Steam turbine produce work using Rankine cycle. In gas turbine, gas enter in compressor and compressor compress the air and reduce the volume of gas and increase the pressure of gas after that gas enters into the combustion chamber where is gas burn and high pressure and high temperature gas expand on gas turbine and turbine convert mechanical energy in to electrical energy. Combined cycle operation employs a heat recovery steam generator (HRSG) that captures heat from high temperature exhaust gases to produce steam, which is then supplied to a steam turbine to generate additional electric power. HRSG is basically a heat exchanger and also called boiler. In which water boil and create steam for steam turbine. Then hot steam expands on steam turbine and steam turbine convert mechanical energy into electrical energy. The valuable quantity of heat is lost during the coupling of two different power or cooling generation unit and its significantly affect the plant performance. The 1st Law of Thermodynamic is treat the work and heat interaction with equivalent forms of energy between system and surrounding. It unable to give the information of thermal deficiency due to internal losses. The assessment of real performance of thermal system and quality of energy are possible by the approach of exergy evaluation. The term Exergy expresses the maximum achievable work or useful quantity obtained from a system at a given state when interacting with environment. It clearly indicates the deficiencies of system due to degradation of energy by process.

2.2 Reviews

2.2.1 A. Khaliq and S et al [1] The aim of the present paper is to use the second-law approach for the thermodynamic analysis of the reheat combined Brayton/Rankine power cycle. Expressions involving the variables for specific power-output, thermal efficiency, exergy destruction in components of the combined cycle, second-law efficiency of each process of the gas-turbine cycle, and second law efficiency of the steam power cycle have been derived. The standard approximation for air with constant properties is used for simplicity. The effects of pressure ratio, cycle temperature ratio, number of reheats and cycle pressure-drop on the combined cycle performance parameters have been investigated. It is found that the exergy destruction in the combustion chamber represents over 50% of the total exergy destruction in the overall cycle. The combined cycle efficiency and its power output were maximized at an intermediate pressure-ratio, and increased sharply up to two reheat-stages and more slowly thereafter.

A development in the search for higher thermal-efficiency of conventional power plant has been the introduction of combined-cycle plants. This is leading to the development of gas turbines dedicated to combined-cycle applications, which has been a subject of great interest in recent years, because of their relatively low initial costs, and the short time needed for their construction. An optimum system for a given power-generation duty may involve alternate cycle configurations, such as compressor intercooling, turbine reheat, and steam injection into the gas turbine combustor.

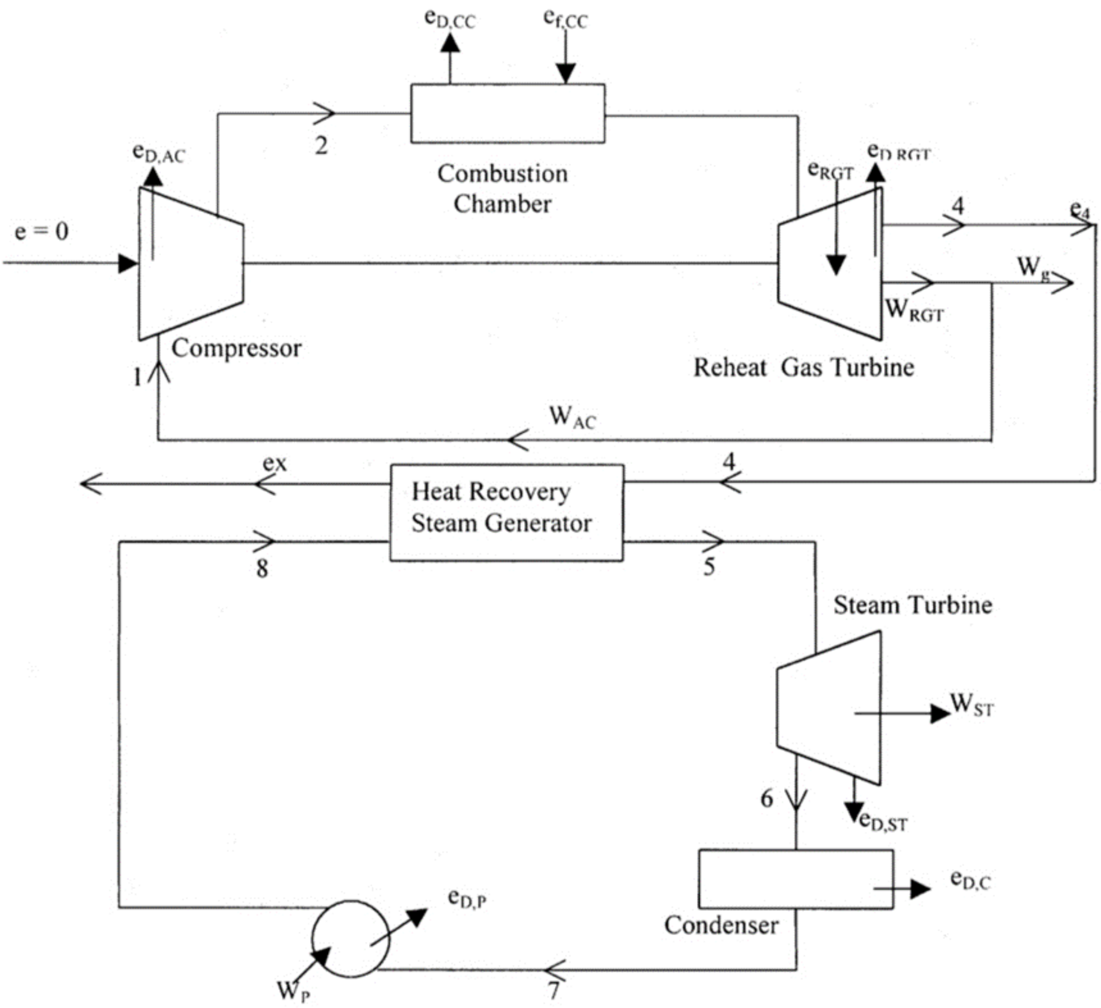

Fig. 4.1 Schematic diagram of the combined Brayton/Rankine power cycle with reheat.

2.2.2 Rajesh Kumar et al [2] A combined 1st and 2nd law of thermodynamic analysis on gas turbine trigeneration system carried out in this paper and observe the effect of pressure ratio, process steam pressure , turbine inlet temperature, refrigeration temperature, etc. the performance of cycle described by 1st law efficiency, electrical to thermal ratio, 2nd law efficiency, and exergy destruction of each component. Energy and exergy analyses previously performed by the authors for a single effect absorption refrigeration system have been extended to double effect vapor absorption refrigeration system with the expectation of reducing energy supply as well as an interest in the diversification of the motive power employed by HVAC technologies. The total exergy destruction in the system as a percentage of the exergy input from a generator heating water over a range of operating temperatures is examined for a system operating on LiBr–H2O solution. The exergy destruction in each component, the coefficient of performance (COP) and the exergetic COP of the system are determined. It is shown that exergy destructions occur significantly in generators, absorbers, evaporator2 and heat exchangers while the exergy destructions in condenser1, evaporator1, throttling valves, and expansion valves are relatively smaller within the range of 1–5%. The results further indicate that with an increase in the generator1 temperature the COP and ECOP increase, but there is a significant reduction in total exergy destruction of the system for the same. On the other hand, the COP and ECOP decrease with an increase in the absorber1 temperature while the total exergy destruction of the system increases significantly with a small increase in the absorber1 temperature. The results show that the exergy method can be used as an effective criterion in designing an irreversible double effect absorption refrigeration system and may be a good tool for the determination of the optimum working conditions of such systems.

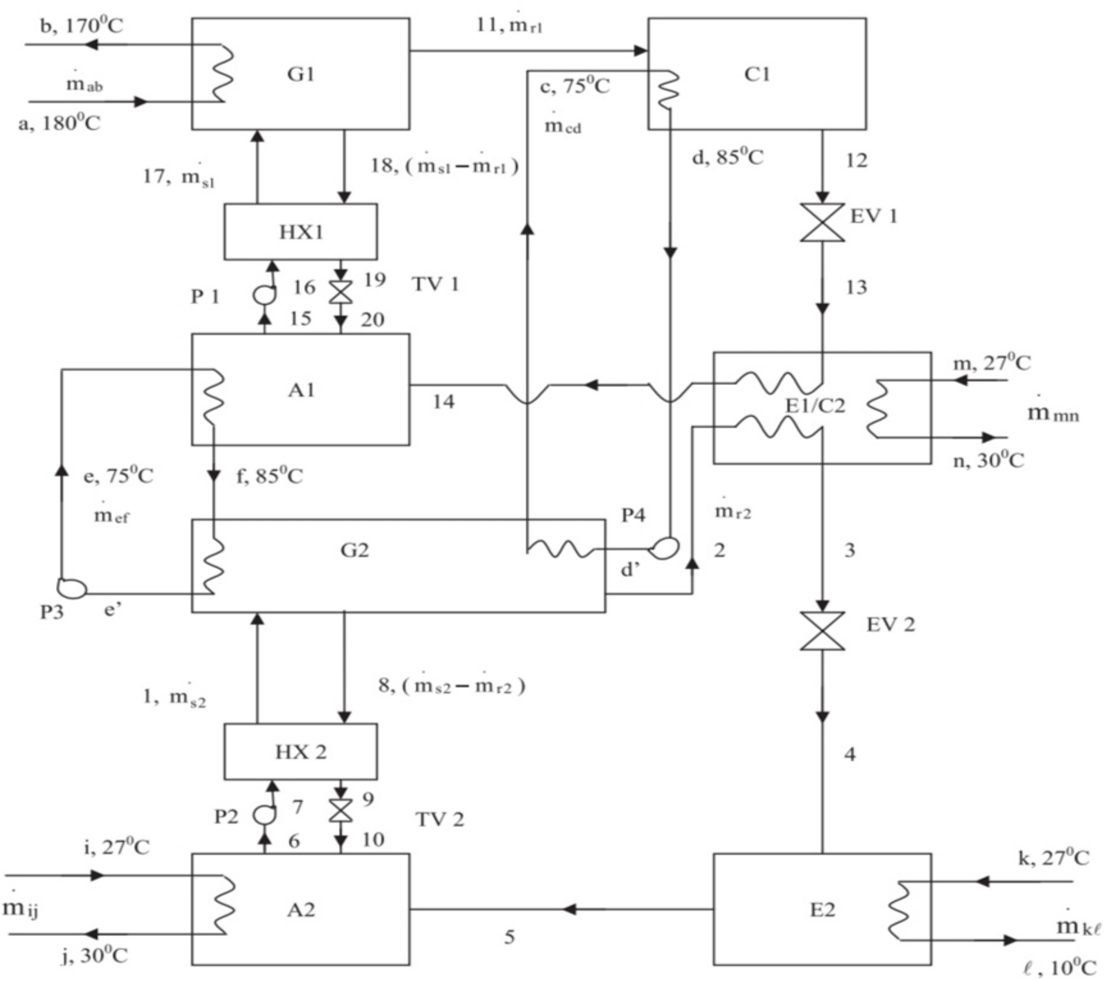

Fig.4.2 Schematic diagram of double effect vapor absorption refrigeration system.

2.2.3 Omendra Singh et al [3] This paper presents thermoeconomic analysis and optimization of a Brayton–Rankine–Kalina combined triple power cycle using Specific Exergy Costing (SPECO) methodology. Cost-balance and auxiliary equations are formulated for each component and for each node and solved through a MATLAB program to get the average cost per unit exergy at different state points. To evaluate the cost effectiveness of the system, the values of thermoeconomic variables for each component are calculated. Large relative cost difference is observed in the steam turbine, HRSG's, combustion chambers, compressors, recuperators and ammonia–water evaporator. Therefore, these components require greater attention. The performance of steam turbine, combustion chambers, recuperators and ammonia–water evaporator can be appreciably improved by capital investment into more efficient design due to their low values of exergoeconomic factor. The performance of HRSG's can be improved only marginally due to slightly higher value of exergoeconomic factor but no such recommendation can be made for the compressors which have a quite high value of exergoeconomic factor. The objective function of the thermoeconomic optimization is the minimization of the total cost rate for the whole plant. Its minimum value is found to occur at a gas cycle pressure ratio of around 14. Decreasing inlet air temperature decreases this objective function parameter significantly while increasing relative humidity causes a small decrease in it.

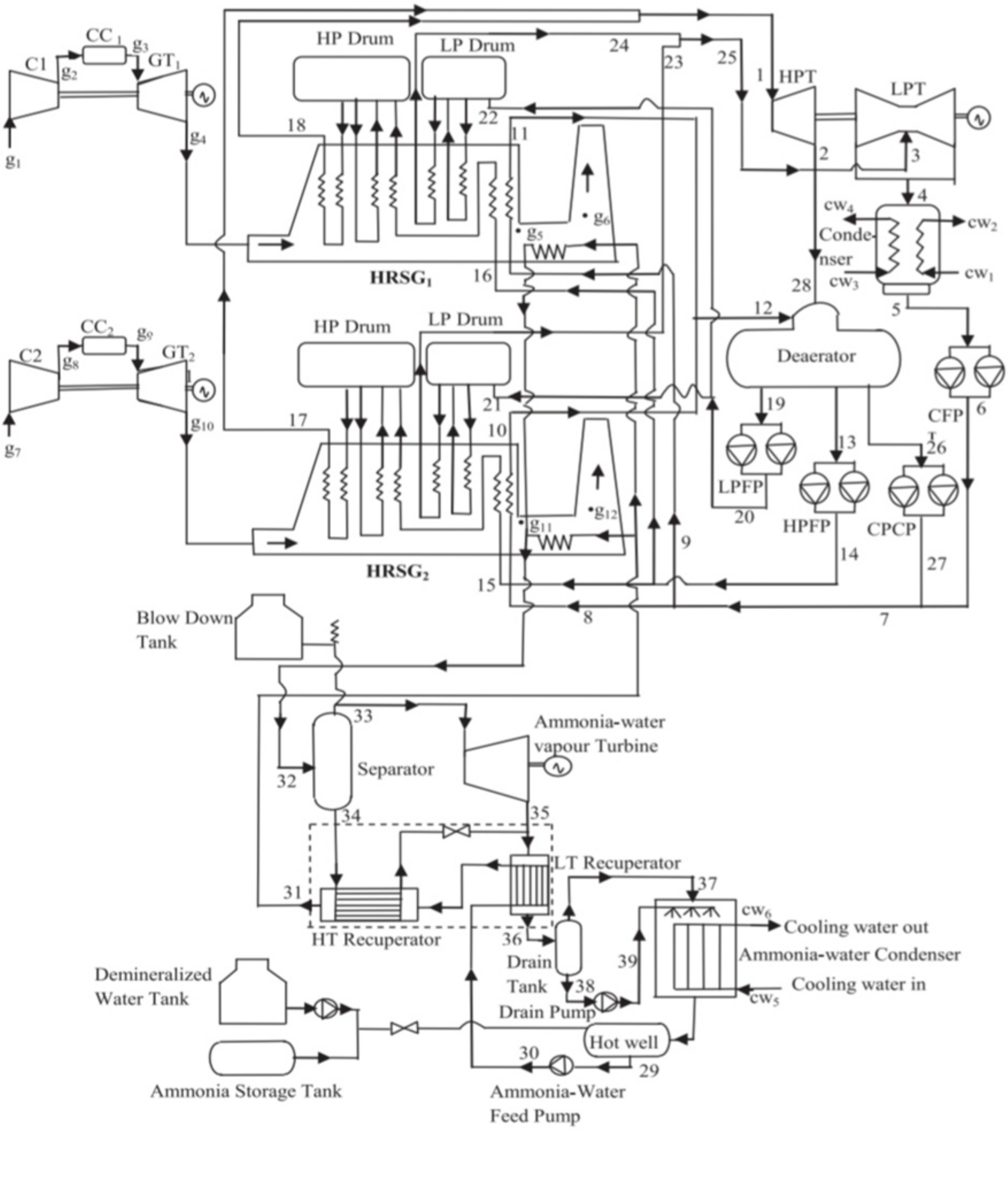

Fig.4.3 Brayton-Rankine-Kalina combined power cycle.

2.2.4 Pouria Ahmadi et al [4] A comprehensive exergy, exergoeconomic and environmental impact analysis and optimization is reported of several combined cycle power plants (CCPPs). In the first part, thermodynamic analyses based on energy and exergy of the CCPPs are performed, and the effect of supplementary firing on the natural gas-fired CCPP is investigated. The latter step includes the effect of supplementary firing on the performance of bottoming cycle and CO2 emissions, and utilizes the first and second laws of thermodynamics. In the second part, a multi-objective optimization is performed to determine the "best" design parameters, accounting for exergetic, economic and environmental factors. The optimization considers three objective functions: CCPP exergy efficiency, total cost rate of the system products and CO2 emissions of the overall plant. The environmental impact in terms of CO2 emissions is integrated with the exergoeconomic objective function as a new objective function. The results of both exergy and exergoeconomic analysis show that the largest exergy destructions occur in the CCPP combustion chamber, and that increasing the gas turbine inlet temperature decreases the CCPP cost of exergy destruction. The optimization results demonstrate that CO2 emissions are reduced by selecting the best components and using a low fuel injection rate into the combustion chamber.

Energy systems involve a large number and various types of interactions with the world outside their physical boundaries. Therefore, designers must address many broad issues, especially energy, economy and the environment. Combined cycle powerplants (CCPPs) have recently received considerable attention due to their relatively high energy efficiencies, low pollutant and greenhouse gas emissions, and operational flexibility. A common CCPP application is the gas-steam combined cycle, which is made up of a gas turbine cycle (topping cycle) and a steam turbine cycle (bottoming cycle) coupled through a heat recovery steam generator (HRSG).To optimize the efficiency, cost effectiveness and environmental impact of such plants, it is important to determine the locations, types and true magnitudes of inefficiencies (irreversibilities). Exergy analysis is a useful tool for such analyses, and permit quantification of the thermodynamic inefficiencies of the process [1-4]. Exergy is the amount of work obtainable for a system as it comes into a state of thermodynamic equilibrium with the surroundings through reversible processes. During the past several decades, many researchers have carried out exergy analyses of power plants.

Dincer and Al-Muslim [5] analyzed a Rankine cycle reheat steam power plant and evaluated the variations in energy and exergy efficiencies at various operating conditions (i.e., boiler temperature and pressure, mass fraction ratio and cycle work output). Rosen and

Dincer [6] studied industrial process heating with steam through exergy analysis, and concluded that exergy analysis should be used as a central tool in process optimization when large quantities of steam are used in energy centers. Via an exergy analysis of supplementary firing in a heat recovery steam generator in a combined cycle power plant, Ameri and Ahmadi [7] showed that energy and exergy efficiencies are reduced if a duct burner is added to a HRSG, but that the CCPP power output increases. Cihan et al.

[8] carried out energy and exergy analysis for a combined cycle located in Turkey and suggested modifications to decrease the exergy destruction in CCPPs. Their results showed that combustion chambers, gas turbines and HRSGs are the main sources of irreversibilities, representing over 85% of the overall exergy losses.

2.2.5 Marc A. Rosen et al [5] The exergy of an energy form or a substance is a measure of its usefulness or quality or potential to cause change. A thorough understanding of exergy and the insights it can provide into the efficiency, environmental impact and sustainability of energy systems, are required for the engineer or scientist working in the area of energy systems and the environment. Further, as energy policies play an increasingly important role in addressing sustainability issues and a broad range of local, regional and global environmental concerns, policy makers also need to appreciate the exergy concept and its ties to these concerns. During the past decade, the need to understand the connections between exergy and energy, sustainable development and environmental impact has become increasingly significant. In this paper, a study of these connections is presented in order to provide to those involved in energy and environment studies, useful insights and direction for analyzing and solving environmental problems of varying complexity using the exergy concept. The results suggest that exergy provides the basis for an effective measure of the potential of a substance or energy form to impact the environment and appears to be a critical consideration in achieving sustainable development.

2.2.6 Marc. A. Rosen et al [6] The relation is investigated between capital costs and thermodynamic losses for devices in modern coalfired, oil-fired and nuclear electrical generating stations. Thermodynamic loss rate-to-capital cost ratios are used to show that, for station devices and the overall station, a systematic correlation appears to exist between capital cost and exergy loss (total or internal), but not between capital cost and energy loss or external exergy loss. The possible existence is indicated of a correlation between the mean thermodynamic loss rate-to-capital cost ratios for all of the devices in a station and the ratios for the overall station, when the ratio is based on total or internal exergy losses. This correlation may imply that devices in successful electrical generating stations are configured so as to achieve an overall optimal design, by appropriately balancing the thermodynamic (exergy-based) and economic characteristics of the overall station and its devices. The results may (i) provide useful insights into the relations between thermodynamics and economics, both in general and for electrical generating stations, (ii) help demonstrate the merits of second-law analysis, and (iii) extend throughout the electrical utility sector.

In the analysis and design of energy systems, techniques are often used which combine scientific disciplines (mainly thermodynamics) with economic disciplines (mainly cost accounting) to achieve optimum designs. For energy conversion devices, cost accounting conventionally considers unit costs based on energy. Many researchers e.g., [1–26] have recommended that costs are better distributed among outputs if cost accounting is based on the thermodynamic quantity exergy. One rationale for this statement is that exergy, but not energy, is often a consistent measure of economic value. In addition, exergy-based economic analysis methodologies exist(e.g., exergoeconomics, thermoeconomics). In previous papers [8,25,26], an alternative approach for discussing the merits of thermoeconomics by identifying as important the ratio of thermodynamic loss rate-to-capital cost was presented in detail. The approach involved examining data for devices in a modern coal-fired electrical generating station, and showing that correlations exist between capital costs and specific second-law-based thermodynamic losses (i.e., total and internal exergy losses). The existence of such correlations likely implies that designers knowingly or unknowingly incorporate into their work the recommendations of exergy analysis.

2.2.7 P. Regulagadda et al [7] In this paper, a thermodynamic analysis of a subcritical boiler–turbine generator is performed for a 32 MW coal-fired power plant. Both energy and exergy formulations are developed for the system. A parametric study is conducted for the plant under various operating conditions, including different operating pressures, temperatures and flow rates, in order to determine the parameters that maximize plant performance. The exergy loss distribution indicates that boiler and turbine irreversibilities yield the highest exergy losses in the power plant. In addition, an environmental impact and sustainability analysis are performed and presented, with respect to exergy losses within the system.

The world energy needs rely heavily on fossil fuels for electricity generation. The majority of the world's power generation is met by fossil fuels, particularly coal and natural gas. Despite the growth of renewable energy installations like wind and solar power, the heavy dependence on fossil fuels is expected to continue for decades. Despite the depletion of fossil fuel reserves and environmental concerns such as climate change, the growth in oil demand is expected to be 47.5% between 2003 and 2030, 91.6% for natural gas and 94.7% for coal [1]. Even though cleaner renewable sources of energy are being rapidly developed, their relative cost and current state of technology have not advanced to a stage where they can significantly reduce our dependence on fossil fuels. Therefore, given the continued reliance on fossil fuels for some time, it is important that fossil fuel plants reduce their environmental impact by operating more efficiently. As energy analysis is based on the first law of thermodynamics, it has some inherent limitations like not accounting for properties of the system environment, or degradation of the energy quality through dissipative processes. An energy analysis does not characterize the irreversibility of processes within the system. In contrast, exergy analysis will characterize the work potential of a system. Exergy is the maximum work that can be obtained from the system, when its state is brought to the reference or ''dead state" (standard atmospheric conditions). Exergy analysis is based on the second law of thermodynamics. This paper will examine a detailed exergy analysis of a thermal power plant, in order to assess the distribution of irreversibilities and losses, which contribute to loss of efficiency in system performance.

2.2.8 Ibrahim Dincer et al [8] In this study, a thermodynamic analysis of a Rankine cycle reheat steam power plant is conducted, in terms of the 1st law of thermodynamic analysis (i.e. energy analysis) and the 2nd law analysis (i.e. exergy analysis), using a spreadsheet calculation technique. The energy and exergy efficiencies are studied as 120 cases for different system parameters such as boiler temperature, boiler pressure, mass fraction ratio and work output. The temperature and pressure values are selected in the range between 400 and 5903C, and 10 and 15 MPa, being consistent with the actual values. The calculated energy and exergy efficiencies are compared with the actual data and the literature work, and good agreement is found. The possibilities to further improve the plant efficiency and hence reduce the inefficiencies are identified and exploited. The results show how exergy analysis can help to make optimum design. The general energy supply and environmental situation requires an improved utilization of energy sources. Therefore, the complexity of power-generating units has increased considerably. Plant owners are increasingly demanding a strictly guaranteed performance. This requires thermodynamic calculations of high accuracy. As a result, the expenditure for thermodynamic calculation during design and optimization has grown tremendously. To be competitive, constructors are forced to reduce planning and designing time as well as the number of errors by applying computer-aided methods. Moreover, the application of computer-aided methods allows optimization and case studies, which otherwise would be too time consuming (Perz, 1991). During the past two decades increasing energy prices and environmental impact has brought the energy issues to the forefront and considerable attention has been paid to efficient energy utilization and process improvement studies and programs. This, in fact, requires accurate thermodynamic analysis of thermal systems for design and optimization purposes. In this regard, there are two essential tools available such as energy analysis (referring to the 1st law of thermodynamic analysis) and exergy analysis (referring to the second law of thermodynamic analysis). Recently, exergy analysis has become a key aspect in providing a better understanding of the process, to quantify sources of inefficiency, to distinguish quality of energy (or heat) used (Jin et al., 1997; Gong and Wall, 1997; Rosen and Dincer, 1997; Dincer and Rosen, 1999).

3

Problem description

3.1 Problem description

Existing system of power plant work under single natural gas in combustion chamber. Its work in a single hydrocarbon gas like ethane or methane or propane or butane. The efficiency of these power plant are very less . The proposed title of work focus on the complete thermal (Performance parameter and energy losses) analysis by using different combustion gases and its different operating temperature and pressure condition and analysis of best suitable operating criteria useful gas with the help of DOE (Design of experiment) approach.

4

THERMODYNAMIC ANALYSIS

4.1 PLANT OPERATION CONDITION-

PLANT COMPONENTS/PARAMETERS	UNIT & VALUE	PLANT COMPONENTS/PARAMETERS	UNIT & VALUE
Compressor Inlet condition (P1, T1)	1bar,25 ^{0}C	Specific heat ratio of air ($\gamma_{\_air}$)	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 ($\gamma_{\_gas}$)	1.33		

4.2 Thermodynamic Analysis of Combined GT-ST

All equations of analysis are based on fundamental approach of first law and second law of thermodynamics, and steam properties as well. A simplified mathematical model of basic thermodynamic approaches is used in analysis. Mass and energy balance equations has been applied in all thermal utilities. In order to simplify the analysis, some assumptions are generally made as follows and adopted from PK Nag GT-ST combined problem analysis [36]:

1. The process is considering steady flow throughout working of system and thermal utilities also consider as a control volume (CV).

2. The state of the mass at every point within the control volume (CV) does not vary with time.

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

4. The thermodynamic equilibrium exits in all units at any given time.

The thermodynamic relation and equations are as follows-

Thermodynamics of GT Plant-

Applying mass-energy and 1st law energy equation in all utility of GT plant

Compressor

The compressor work rate is a function of air mass flow rate of air, specific heat, isentropic efficiency of compressor and temperature difference, the outlet temperature can be expressed as follows:

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

$$T2/T1=([P2/P1])^{((\gamma-1))/((\gamma * \eta c))} \quad (1)$$

$$W\_comp= m\_air \times Cp\_air (T2-T1) \quad (1a)$$

Combustor

The combustion chamber outlet pressure is defined by considering a pressure drop (ΔPcc) across the combustion chamber as follows

$$P3/P2 = (1- \Delta Pcc) \quad (2)$$

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 $f \times CVf = mf \times Cp\_gas\,(T3-T4) - (1-f)\,Cp\_air\,(T2-T0)$ (4)

Air Fuel Ratio = $A/F = ((1-f))/f$ (5)

Octane Combustion reaction occurs as follows

$C8H8 + 12.5\,O2 = 8CO2 + 9H20$ (6)

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

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

Gas Turbine

The gas turbine outlet temperature can be written a function of gas turbine isentropic efficiency (ηGT), the gas turbine inlet temperature (T3) and gas turbine pressure ratio (P3/P4) as follows:

P4 = 1.05 bar, T3 = 900 0C

$T3/T4 = (⟦P3/P4)⟧\hat{}(((\gamma-1)(\eta GT))/((\gamma)))$ (7)

Heat Recovery Steam Generator (HRSG)

By applying the energy balance for gas and water in each part of the HRSG, the gas temperature and water properties are calculated in each part of the HRSG by considering pinch point temperature difference between gas turbine exist temperature and saturation temperature of HRSG working pressure as following:

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 = Tf+ 30

From steam table at 40 bar –at Ta =ST inlet temp=425 0C = ha=3272 KJ/Kg

hf=1087 KJ/Kg (at Tf)

From Mass and Energy balance between GT and ST

$$m\_gas \times Cp\_gas\ (T4-T5) = m\_steam\ (ha-hf) \quad (8)$$

If 1 Kg/s of gas flow in GT, then steam flow rate is in terms of per kg of gas flow

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

And air flow rate entering the compressor = m_air = (1-f) m_gas (9)

Fuel mass flow rate m_fuel =wf =f× m_gas (9a)

The heat added (he) in the HRSG through Feed water temp (Te) at 170.4 0C

he=721.1 KJ/Kg

Then energy interaction between HRSG and stack flow, and temperature of stack flue gas as follows

$$1.14\ (482-T6) = 0.106\ (3272-721.1) \quad (9b)$$

Stack flow Temp= T6

Power Output Of GT-ST Plant-

The combined GT-ST plant power output is achieved by the net work done by gas turbine and steam turbine both, total work of GT& ST plant is given by

W_total = W_ST plant + W_GTplant

But the steam turbine work done is a function of isentropic efficiency of steam turbine, which is given already, so thermal equation for steam turbine working terms of enthalpy across inlet and outlet of ST as follows

$$W\_ST = ws\,(ha - hbs) \times \eta\_st \qquad (10)$$

From enthalpy and entropy equation between steam turbine expansion process

$$h = hf + xhfg \qquad (11)$$

$$s = sf + x\,sfg \qquad (12)$$

At 40 bar-425 0C at inlet of ST and 0.04 bar at condenser line enthalpies are as follows

Enthalpy, ha = 3272 KJ/Kg and Entropy sa = 6.853 KJ/Kg-K from steam property table

But after expansion of steam, steam quality also occurs in terms of dryness fraction, and entropy also changed as follows

$$\text{So, } sbs = sf\_b + x\_bs \times sfg\_b \qquad (13)$$

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

sa = 6.853 KJ/Kg-K = sbs = 0.4226 +x_bs × 8.052

Steam quality after expansion = x_bs = 0.7986 or 80%

$$\text{And Enthalpy at condenser line} = hbs = hf\_b + x\_bs \times hfg\_b \qquad (14)$$

Work done by gas turbine plant is depend on the difference between gas turbine work and compressor work consumption, the combine equation for W_GT and W_COMP as follows

$$W\_GT\ plant = W\_GT - W\_COMP \qquad (15)$$

$$W\_GT\ plant = m\_gas \times Cp\_gas\,(T3-T4) - m\_air \times Cp\_air\,(T2-T1) \qquad (15a)$$

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

$$XL = ((wg \times Cpg(T6-T1))/(wf \times CVf) \quad (16)$$

For the overall performance in terms of the energy efficiencies for the topping (gas turbine) cycle, bottoming (steam turbine) cycle and overall plant respectively, are evaluated with considering heat lost between topping and bottoming cycle the overall efficiency of plant as follows

$$\eta overall\ plant = \eta STplant + \eta GTplant - \eta STplant \times \eta GTplant - \eta STplant \times XL \quad (17)$$

The plant efficiency of GT & ST plant is depending on work done by both turbines and heat supplied through combustion chamber and HRSG respectively in GT and ST plant. The equations for efficiencies are follows

$$\eta STplant = (ha-hb)/(ha-he) \quad (18)$$

put all values of entropy from equ. No 10-11-12-13

$$\eta GTplant = (WGT)/(wf \times CVf) \quad (19)$$

From Equ No 17 ηoverall

Exergy Analysis

Assume exergy flux $\psi = (\Delta G0/ \Delta H0) = 1.0401 + 0.1728\ (h/c)$

(20)

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

$\psi = (\Delta G0/ \Delta H0) = 1.0401 + 0.1728\ (18 \times 1/8 \times 12)$

$\Delta H0 = wf \times (CV)0$ (21)

From Equ no 20

Exergy Input $=\Delta G0= \psi \times \Delta H0$

$T0\Delta S0 = \Delta G0-\Delta H0$ (22)

Exergy destruction of components due to irreversibility in process, the all equations of irreversibility are in terms of TdS form, where temperature T is ambient temperature as T0

Compressor-

Rate of energy dissipation in compressor (Icomp) =waT0 (s2-s1) (23)

Now, $s2-s1=Cpa \ln T2/T1- Ra \ln P2/P1$

(24)

But $Ra = Cp ((\gamma-1)/\gamma)$ (25)

Combustion Chamber-

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

$Icc = T0 [(SP)3-(SR)2$ (26)

$Icc = T0 [\{(SP)3-(SP) O\} +\{(SA)2-(SA) O\} +\Delta S0$ (26a)

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

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

Gas Turbine-

Rate of energy lost or work lost in GT = IGT =wgT0 (s4-s3) (27)

Where s4-s3 = Cpg $\ln(T4/T3)$-Rg $\ln(p4/p3)$ (28)

IGT =wgT0 [Cpg $\ln(T4/T3)$-Rg $\ln(p4/p3)$] (30)

HRSG-

Rate of energy lost in heat recovery steam generator-IHRSG

IHRSG = T0[ws (Sa-Se) +wg(s6-s4)] (31)

IHRSG = T0[ws(sa-se) +Cpg ln T6/T4-Rg ln P6/P4] (31a)

Steam Turbine

Rate of energy or work lost in the steam turbine= IST

IST = T0 ws (sb-sa) (32)

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

For sb = sb= (sf +x sfg)b (33)

But steam quality at state 'b'

hb = (hf+xhfg)b (34)

but ha-hb = ηST (ha-hbs) (35)

From Steam properties table-

sa = 6.853 kJ/kg. -K, ha = 3272 kJ/kg., hbs = 2064.37kJ/kg.

And ηST = 82% = 0.82 (taken data)

Put all values in equation no (35)

xb = 0.89

Exergy Lost due to exhaust flue gas

$$IEXHFLUEGAS = \int_{T0}^{T6} (1-T0/T)\, dQ = wg \times Cpg\, [(T6-T0)-T0 \ln T6/T0] \quad (36)$$

$$\text{Exergetic Efficiency} = \eta EX = \text{(Total Output)/ (Exergy Input)} \quad (37)$$

5

RESULT & CONCLUSION

5.1 Result

	METHANE				
S. No	Parameters	OPERATING CONDITION-COMPRESSION RATIO (CR=P2/P1) and T3 (GTIT IN DEGREE C)			
		CR=8, T3=900	CR=10, T3=800	CR=12, T3=700	CR=15, T3=600
1	T4 (GTET)	485	388	303	
2	A/F	32.4046511	38.78773148	47.94678055	
3	T5 (GT EXHAUST FLOW TEMP)	280	280	280	
4	m_gas	144.0342782	273.3983984	1283.783784	
5	m_air	139.7224751	266.5269738	1257.555628	
6	m_fuel	4.311803102	6.871424638	26.22815576	
7	W_GT	92.21726724	161.0432341	663.3626443	
8	η_GT	0.38535442	0.422282139	0.4557118	
9	XL	0.30064186	0.358089583	0.440521025	
10	η_overall	0.504675834	0.505740885	0.495143327	
11	Ψ	1.0977	1.0977	1.0977	
12	ΔHo	239305.0722	381364.0674	1455662.645	
13	ToΔSo	23380.10555	37259.26939	142218.2404	
14	I_compressor	3438.938876	7250.659872	36652.46058	
15	I_combustion chamber	65952.23254	84169.86379	212074.2988	
16	I_GT	5684.061866	12153.49068	64714.40114	

	ETHANE				
S. No	Parameters	OPERATING CONDITION-COMPRESSION RATIO (CR=P2/P1) and T3 (GTIT IN DEGREE C)			
		CR=8, T3=900	CR=10, T3=800	CR=12, T3=700	CR=15, T3=600
1	T4 (GTET)	485	388	303	
2	A/F	48.19846486	59.54067939	77.00974975	
3	T5 (GT EXHAUST FLOW TEMP)	280	280	280	
4	m_gas	208.9909134	396.6957153	1862.745098	
5	m_air	204.7429981	390.1431672	1838.86675	
6	m_fuel	4.247915336	6.552548127	23.87834808	
7	W_GT	73.37900111	119.7564144	446.9870183	
8	η_GT	0.332834691	0.352144741	0.360681007	
9	XL	0.326330858	0.401563177	0.517434928	
10	η_overall	0.462465782	0.445735732	0.407076971	
11	Ψ	1.0833	1.0833	1.0833	
12	ΔHo	220446.8059	340077.2478	1239286.265	
13	ToΔSo	18364.88493	28328.43474	103232.5459	
14	I_compressor	5039.265554	10613.54266	53595.23632	
15	I_combustion chamber	72517.94026	99619.22864	296815.7263	
16	I_GT	5684.061864	12153.49068	64714.40112	

	PROPANE				
S. No	parameters	OPERATING CONDITION-COMPRESSION RATIO (CR=P2/P1) and T3 (GTIT IN DEGREE C)			
		CR=8, T3=900	CR=10, T3=800	CR=12, T3=700	CR=15, T3=600
1	T4 (GTET)	485	388	303	
2	A/F	41.77365	51.13807956	65.25932744	
3	T5 (GT EXHAUST FLOW TEMP)	280	280	280	
4	m_gas	191.4707171	363.4397871	1706.586826	
5	m_air	186.9943463	356.4690703	1680.830651	
6	m_fuel	4.476370782	6.9707168	25.7561749	
7	W_GT	78.52254177	131.0032934	505.8112149	
8	η_GT	0.348046892	0.372884421	0.389651679	
9	XL	0.318892837	0.388708004	0.49398695	
10	η_overall	0.474609795	0.4634793	0.433869	
11	Ψ	1.0785	1.0785	1.0785	
12	ΔHo	225609.0874	351324.1267	1298111.215	

13	ToΔSo	17710.31336	27578.94395	101901.7304	
14	I_compressor	4602.424389	9697.46494	48989.14832	
15	I_combustion chamber	68702.14589	92228.36524	261848.273	
16	I_GT	5684061865	12153.49068	64114.40111	

	BUTANE				
S. No	parameters	**OPERATING CONDITION-COMPRESSION RATIO (CR=P2/P1) and T3 (GTIT IN DEGREE C)**			
		CR=8, T3=900	CR=10, T3=800	CR=12, T3=700	CR=15, T3=600
1	T4 (GTET)	485	388	303	
2	A/F	40.66398241	49.78427673	63.53754155	
3	T5 (GT EXHAUST FLOW TEMP)	280	280	280	
4	m_gas	191.4707171	363.4397871	1706.586826	
5	m_air	186.8751238	356.2832455	1680.143507	
6	m_fuel	4.595593269	7.156541562	26.44331943	
7	W_GT	78.55582462	131.0653574	506.0669836	
8	η_GT	0.348141114	0.372995186	0.389771913	
9	XL	0.31884402	0.388639348	0.493889638	
10	η_overall	0.474686763	0.463574063	0.433980524	
11	Ψ	1.0761	1.0761	1.0761	
12	ΔHo	225643.6295	351386.1907	1298366.984	
13	ToΔSo	17171.48021	26740.48911	98805.72748	
14	I_compressor	4599.490009	9692.40972	48969.12096	
15	I_combustion chamber	68142.05466	91353.26111	258606.0173	
16	I_GT	5684.061865	12153.49068	64714.40111	

	PENTANE				
S. No	parameters	**OPERATING CONDITION-COMPRESSION RATIO (CR=P2/P1) and T3 (GTIT IN DEGREE C)**			
		CR=8, T3=900	CR=10, T3=800	CR=12, T3=700	CR=15, T3=600
1	T4 (GTET)	485	388	303	
2	A/F	40.547502	49.67254868	63.45151548	
3	T5 (GT EXHAUST FLOW TEMP)	280	280	280	
4	m_gas	192.6241552	365.6291834	1716.86747	
5	m_air	187.9879161	358.4136556	1690.229346	
6	m_fuel	4.636239145	7.215527794	26.63812413	
7	W_GT	78.22341756	130.3538174	502.3128329	
8	η_GT	0.347208384	0.371722955	0.388002359	
9	XL	0.319300247	0.38942792	0.495321832	
10	η_overall	0.473935104	0.462485622	0.432339166	

11	Ψ	1.07466	1.07466	1.07466	
12	ΔHo	225321.2224	350674.6508	1294612.833	
13	ToΔSo	16822.48247	26181.36943	96655.79409	
14	I_compressor	4626.878765	9750.365877	49263.08077	
15	I_combustion chamber	67991.25846	91214.31149	258602.7688	
16	I_GT	5684.061867	12153.49608	64714.40113	

	OCTANE				
S. No	Parameters	**OPERATING CONDITION-COMPRESSION RATIO (CR=P2/P1) and T3 (GTIT IN DEGREE C)**			
		CR=8, T3=900	CR=10, T3=800	CR=12, T3=700	CR=15, T3=600
1	T4 (GTET)	485	388	303	
2	A/F	35.53336227	43.41990135	55.25079692	
3	T5 (GT EXHAUST FLOW TEMP)	280	280	280	
4	m_gas	186.8825819	354.7308267	1665.692577	
5	m_air	181.7671868	346.7449732	1636.080684	
6	m_fuel	5.115395088	7.985853545	29.61189296	
7	W_GT	80.03573702	134.251064	522.4680475	
8	η_GT	0.352388556	0.378628608	0.397384199	
9	XL	0.316766467	0.385147557	0.487728616	
10	η_overall	0.478109647	0.468393665	0.441041329	
11	Ψ	1.0725	1.0725	1.0725	
12	ΔHo	227123.5419	354571.8974	1314768.047	
13	ToΔSo	16466.45679	25706.46256	95320.68344	
14	I_compressor	4473.770198	9432.92841	47684.87488	
15	I_combustion chamber	66527.24692	88438.04913	245742.5766	
16	I_GT	5684.061867	12153.49068	64714.40111	

5.2 Conclusion

It has been found, the Methane gas has performed in terms of better efficiency, least heat loss and more turbine work at higher compression ratio and 800^0 C of inlet temperature of gas turbine. So for combined power plant, methane gas is the most suitable and efficient hydrocarbon fuel.

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