

A Project Report On

**STREAMLINING AND THE IMPACT OF STEAM TURBINE OUTLET
QUALITY ON THE YIELD FORCE OF A JOINED CYCLE POWER
PLANT**

Submitted in partial fulfillment of the
Requirements for the award of the degree of

MASTER OF TECHNOLOGY

**In
MECHANICAL ENGINEERING**

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PYDAH COLLEGE OF ENGINEERING

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CERTIFICATE

This is to certify that the project titled “**STREAMLINING AND THE IMPACT OF STEAM TURBINE OUTLET QUALITY ON THE YIELD FORCE OF A JOINED CYCLE POWER PLANT**” to internal guide **Mr. G. SREENIVAS, M.Tech.,(Ph.d)**., Mechanical Engineering, in partial fulfillment of the requirements for the award of the degree of **MASTER OF TECHNOLOGY in MECHANICAL ENGINEERING** from Pydah College Of Engineering, Patavala.

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DECLARATION

This is to certify that the project titled “**Streamlining and the impact of steam turbine outlet quality on the yield force of a joined cycle power plant**” is a bonafide work done by me, in partial fulfillment of the requirements for the award of the degree M.Tech (Thermal engineering) and submitted to the Department of Mechanical Engineering, **Pydah College of Engineering, Jawaharlal Nehru Technological University, Kakinada**. This is a bonafide original work carried out by me under the guidance and supervision of Mr. **G.SREENIVAS, M.Tech, (Ph.d)**, our faculty, Pydah College of Engineering, Kakinada. I also declare that this project is a result of my own effort and that has not been copied from anyone and have taken only citations from the sources which are mentioned in the references. This work was not submitted earlier at any other University or Institute for the award of any degree.

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ABSTRACT

A restricted way exists to a maintainable arrangement which goes through watchful strides of efficiency change (asset administration) and gives natural amicable energies. Warm power plants are more regular in numerous force generation destinations around the globe. Consequently, in this ebb and flow examination concentrate on a thorough thermodynamic demonstrating of a consolidated combined cycle plant with double weight heat recuperation steam generator is displayed. Since the steam turbine outlet quality is prohibitive parameters, advancement of three cases with distinctive steam quality are led and talked about. In other hand, vitality and exergy investigation of every segments for these three distinct cases evaluated and thought about. Gotten results demonstrate that it is truly essential to keep the nature of the vapor at turbine outlet steady in 88% for the outcomes to be more reasonable furthermore advancement and information are all the more in fact achievable and appropriate.

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Introduction

Combined Cycle

In electric power generation a combined cycle is an assembly of heat engines that work in tandem from the same source of heat, converting it into mechanical energy, which in turn usually drives electrical generators. The principle is that after completing its cycle (in the first engine), the working fluid of the first heat engine is still low enough in its entropy that a second subsequent heat engine may extract energy from the waste heat (energy) of the working fluid of the first engine. By combining these multiple streams of work upon a single mechanical shaft turning an electric generator, the overall net efficiency of the system may be increased by 50 – 60 percent. That is, from an overall efficiency of say 34% (in a single cycle) to possibly an overall efficiency of 51% (in a mechanical combination of two (2) cycles) in net Carnot thermodynamic efficiency. This can be done because heat engines are only able to use a portion of the energy their fuel generates (usually less than 50%). In an ordinary (non combined cycle) heat engine the remaining heat (e.g., hot exhaust fumes) from combustion is generally wasted.

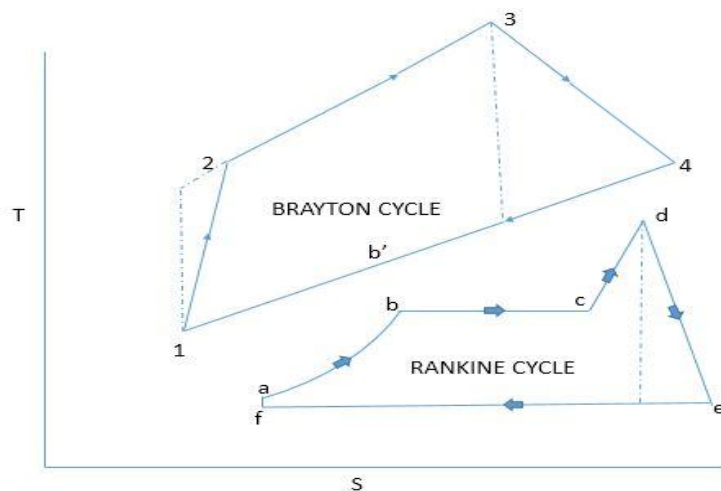
Combining two or more thermodynamic cycle's results in improved overall efficiency, reducing fuel costs. In stationary power plants, a widely used combination is a gas turbine (operating by the Brayton cycle) burning natural gas or synthesis gas from coal, whose hot exhaust powers a steam power plant (operating by the Rankine cycle). This is called a Combined Cycle Gas Turbine (CCGT) plant, and can achieve a best-of-class real (HHV - see below) thermal efficiency of around 54% in base-load operation, in contrast to a single cycle steam power plant which is limited to efficiencies of around 35-42%. Many new gas power plants in North America and Europe are of this type. Such an arrangement is also used for marine propulsion, and is called a combined gas and steam (COGAS) plant. Multiple stage turbine or steam cycles are also common.

Other historically successful combined cycles have used hot cycles with mercury vapor turbines, magneto hydrodynamic generators or molten carbonate fuel cells, with steam plants for the low temperature "bottoming" cycle. However, it is common in cold climates (such as Finland) to drive community heating systems from a power plant's condenser heat. Such cogeneration systems can yield theoretical efficiencies above 95%.

1.2 Basic combined cycle

The thermodynamic cycle of the basic combined cycle consists of two power plant cycles. One is the Joule or Brayton cycle which is a gas turbine cycle and the other is Rankine cycle which is a steam turbine cycle.^[1] The cycle 1-2-3-4-1 which is the gas turbine power plant cycle is the topping cycle. It depicts the heat and work transfer process taking place in high temperature region.

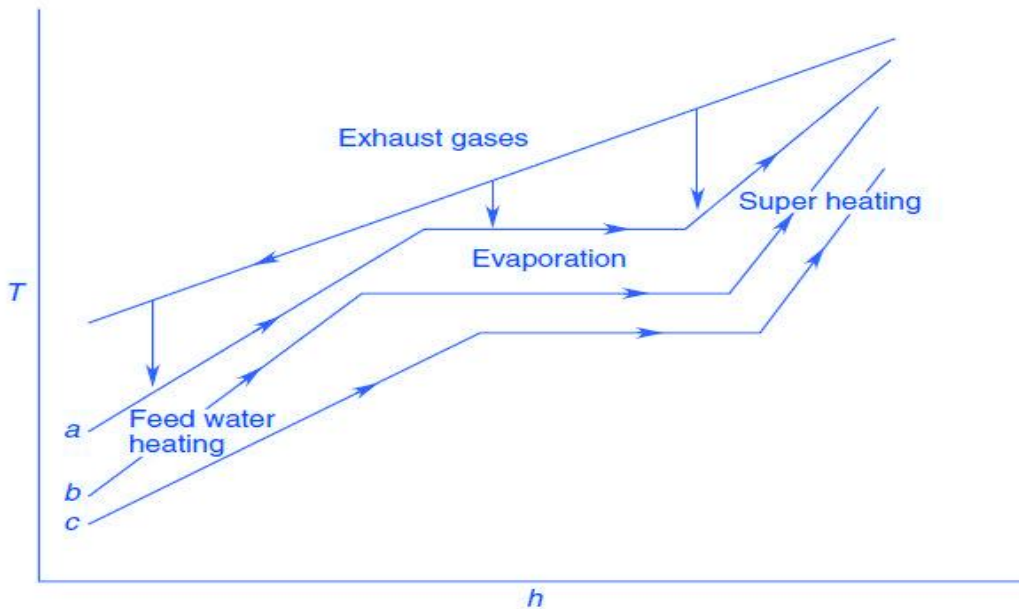
The cycle a-b-c-d-e-f-a which is the Rankine steam cycle takes place at a low temperature and is known as the bottoming cycle. Transfer of heat energy from high temperature exhaust gas to water and steam takes place by a waste heat recovery boiler in the bottoming cycle. During the constant pressure process 4-1 the exhaust gases in the gas turbine reject heat. The feed water, wet and super heated steam absorb some of this heat in the process a-b, b-c and c-d.



1.2.1 Steam generators

The steam power plant gets its input heat from the high temperature exhaust gases from gas turbine power plant.^[1] The steam thus generated thus can be used to drive steam turbine. The Waste Heat Recovery Boiler (WHRB) has 3 sections: Economizer, evaporator and super heater.

Initially the feed water entered in economizer where it can preheated by the waste flue gases leaving the steam generator after that the water enter into steam generator also called as evaporator where it can be heated by flue gases and finally the steam entered into super heater where the steam further heated by the flue gas having high temperature and final steam will leave from the super heater to steam turbine. The flue gas enter into super heater where it has high temperature then it will enter into evaporator and then finally into economizer after it will exhausted to the atmosphere through chimney.



1.3 Design Principle

The efficiency of a heat engine, the fraction of input heat energy that can be converted to useful work, is limited by the temperature difference between the heat entering the engine and the exhaust heat leaving the engine.

In a thermal power station, water is the working medium. High pressure steam requires strong, bulky components. High temperatures require expensive alloys made from nickel or cobalt, rather than inexpensive steel. These alloys limit practical steam temperatures to 655 °C while the lower temperature of a steam plant is fixed by the temperature of the cooling water. With these limits, a steam plant has a fixed upper efficiency of 35% to 42%.

An open circuit gas turbine cycle has a compressor, a combustor and a turbine. For gas turbines the amount of metal that must withstand the high temperatures and pressures is small, and lower quantities of expensive materials can be used. In this type of cycle, the input temperature to the turbine (the firing temperature), is relatively high (900 to 1,400 °C). The output temperature of the flue gas is also high (450 to 650 °C). This is therefore high enough to provide heat for a second cycle which uses steam as the working fluid (a Rankine cycle).

In a combined cycle power plant, the heat of the gas turbine's exhaust is used to generate steam by passing it through a heat recovery steam generator (HRSG) with a live steam temperature between 420 and 580 °C. The condenser of the Rankine cycle is usually

cooled by water from a lake, river, sea or cooling towers. This temperature can be as low as 15 °C.

1.4 Typical size of CCGT plants

For large-scale power generation, a typical set would be a 270 MW gas turbine coupled to a 130 MW steam turbine giving 400 MW. A typical power station might consist of between 1 and 6 such sets.

Plant size is important in the cost of the plant. The larger plant sizes benefit from economies of scale (lower initial cost per kilowatt) and improved efficiency.

Gas turbines of about 150 MW size are already in operation manufactured by at least four separate groups – General Electric and its licensees, Alstom, Siemens, and Westinghouse/Mitsubishi. These groups are also developing, testing and/or marketing gas turbine sizes of about 200 MW. Combined cycle units are made up of one or more such gas turbines, each with a waste heat steam generator arranged to supply steam to a single steam turbine, thus forming a combined cycle block or unit. Typical Combined cycle block sizes offered by three major manufacturers (Alstom, General Electric and Siemens) are roughly in the range of 50 MW to 500 MW and costs are about \$600/kW.

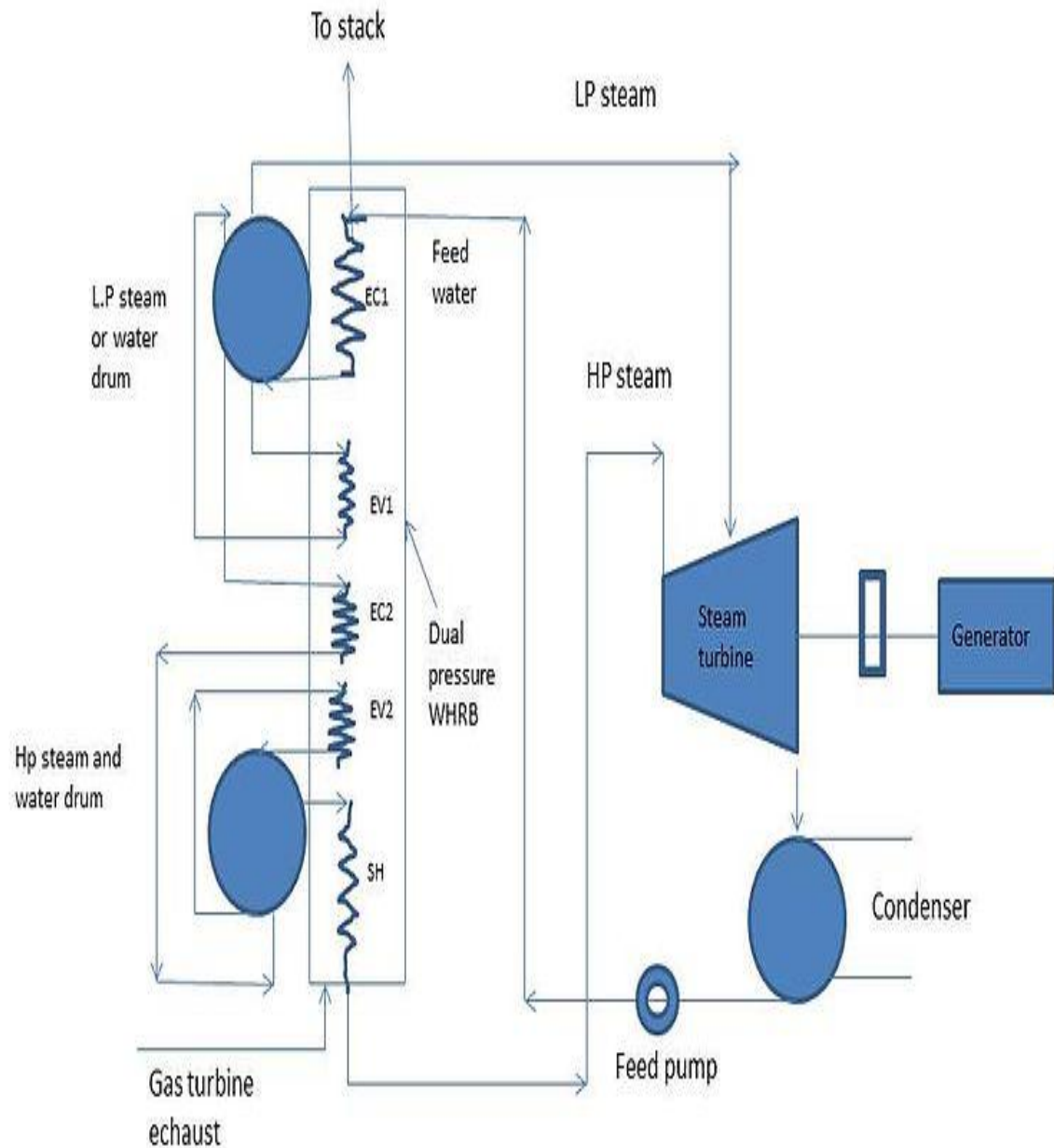
1.4.1 Unfired Boiler

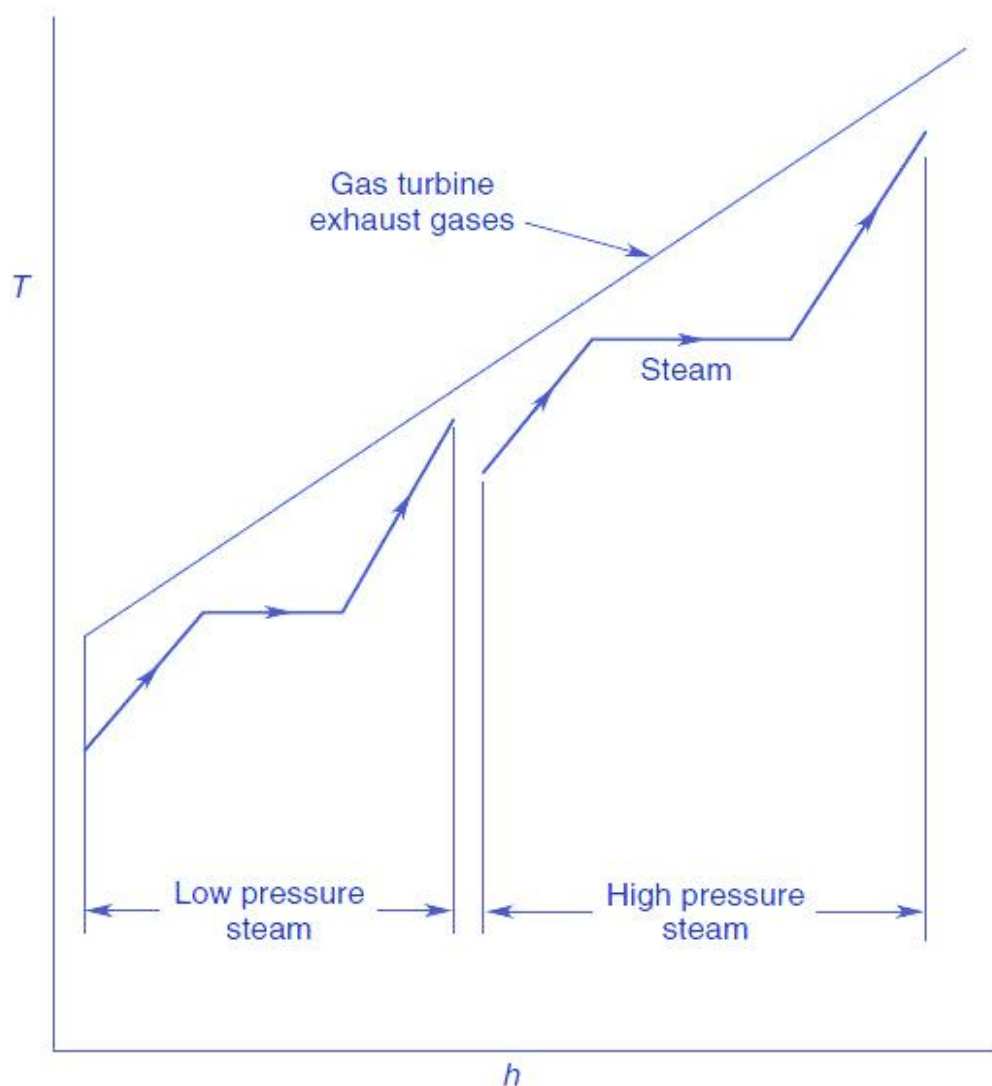
The heat recovery boiler is item 5 in the COGAS figure shown above. No combustion of fuel means that there is no need of fuel handling plant and it is simply a heat exchanger. Exhaust enters in super heater and the evaporator and then in to economiser section as it flows out from the boiler. Feed water comes in through the economizer and then exits after having attained saturation temp in the water or steam circuit. Finally it then flows through evaporator and super heater. If the temperature of the gases entering the heat recovery boiler is higher, then the temperature of the exiting gases is also high.

1.4.2 Dual pressure Boiler

It is often desirable if high heat is recovered from the exiting gases. Hence dual pressure boiler is employed for this purpose. It has two water/steam drums. Low pressure drum is connected to low pressure economizer or evaporator. The low pressure steam is generated in low temperature zone. The low pressure steam is supplied to the low temperature turbine. Super heater can be provided in the low pressure circuit.

Some part of the feed water from the low-pressure zone is transferred to the high-pressure economizer by a booster pump. This economizer heats up the water to its saturation temperature. This saturated water goes through the high-temperature zone of the boiler and is supplied to the high-pressure turbine.





1.4.3 Supplementary firing and blade cooling

Supplementary firing may be used in combined cycles (in the HRSG) raising exhaust temperatures from 600 °C (GT exhaust) to 800 or even 1000 °C. Using supplemental firing will however not raise the combined cycle efficiency for most combined cycles. For single boilers it may raise the efficiency if fired to 700–750 °C; for multiple boilers however, supplementary firing is often used to improve peak power production of the unit, or to enable higher steam production to compensate for failure of a second unit.

Maximum supplementary firing refers to the maximum fuel that can be fired with the oxygen available in the gas turbine exhaust. The steam cycle is conventional with reheat and regeneration. Hot gas turbine exhaust is used as the combustion air. Regenerative air preheater is not required. A fresh air fan which makes it possible to operate the steam plant even when the gas turbine is not in operation, increases the availability of the unit.

The use of large supplementary firing in Combined Cycle Systems with high gas turbine inlet temperatures causes the efficiency to drop. For this reason the Combined Cycle Plants with maximum supplementary firing are only of minimal importance today, in comparison to simple Combined Cycle installations. However, they have two advantages that is a) coal can be burned in the steam generator as the supplementary fuel, b) has very good part load efficiency.

The HRSG can be designed with supplementary firing of fuel after the gas turbine in order to increase the quantity or temperature of the steam generated. Without supplementary firing, the efficiency of the combined cycle power plant is higher, but supplementary firing lets the plant respond to fluctuations of electrical load. Supplementary burners are also called *duct burners*.

More fuel is sometimes added to the turbine's exhaust. This is possible because the turbine exhaust gas (flue gas) still contains some oxygen. Temperature limits at the gas turbine inlet force the turbine to use excess air, above the optimal stoichiometric ratio to burn the fuel. Often in gas turbine designs part of the compressed air flow bypasses the burner and is used to cool the turbine blades.

Supplementary firing raises the temperature of the exhaust gas from 800 to 900 degree Celsius. Relatively high flue gas temperature raises the condition of steam (84 bar, 525 degree Celsius) thereby improving the efficiency of steam cycle.

1.5 Fuel for combined cycle power plants

The turbines used in Combined Cycle Plants are commonly fueled with natural gas. The improvement in shale gas extraction has increased gas supplies and reserves dramatically.^[2] Because of this fact, it is becoming the fuel of choice for an increasing amount of private investors and consumers because it is more versatile than coal or oil and can be used in 90% of energy applications. Chile which once depended on hydro-power for 70% of its electricity supply, is now boosting its gas supplies to reduce reliance on its drought afflicted hydro dams. Similarly China is tapping its gas reserves to reduce reliance on coal, which is currently burned to generate 80% of the country's electricity supply.

Where the extension of a gas pipeline is impractical or cannot be economically justified, electricity needs in remote areas can be met with small-scale Combined Cycle Plants, using renewable fuels. Instead of natural gas, Combined Cycle Plants can be filled with biogas derived from agricultural and forestry waste, which is often readily available in rural areas.

Combined cycle plants are usually powered by natural gas, although fuel oil, synthesis gas or other fuels can be used. The supplementary fuel may be natural gas, fuel oil, or coal. Biofuels can also be used. Integrated solar combined cycle power stations combine the energy harvested from solar radiation with another fuel to cut fuel costs and environmental impact (look ISCC section). Next generation nuclear power plants are also on the drawing board which will take advantage of the higher temperature range made available by the Brayton top cycle, as well as the increase in thermal efficiency offered by a Rankine bottoming cycle.

1.5.1 Low-Grade Fuel for Turbines

Gas turbines burn mainly natural gas and light oil. Crude oil, residual, and some distillates contain corrosive components and as such require fuel treatment equipment. In addition, ash deposits from these fuels result in gas turbine deratings of up to 15 percent. They may still be economically attractive fuels however, particularly in combined-cycle plants.

Sodium and potassium are removed from residual, crude and heavy distillates by a water washing procedure. A simpler and less expensive purification system will do the same job for light crude and light distillates. A magnesium additive system may also be needed to reduce the corrosive effects if vanadium is present. Fuels requiring such treatment must have a separate fuel-treatment plant and a system of accurate fuel monitoring to assure reliable, low-maintenance operation of gas turbines.

1.6 Configuration

A single shaft combined cycle plant comprises a gas turbine and a steam turbine driving a common generator. In a multi-shaft combined cycle plant, each gas turbine and each steam turbine has its own generator. The single shaft design provides slightly less initial cost and slightly better efficiency than if the gas and steam turbines had their own generators. The multi-shaft design enables two or more gas turbines to operate in conjunction with a single steam turbine, which can be more economical than a number of single shaft units.

The primary disadvantage of single shaft combined cycle power plants is that the number of steam turbines, condensers and condensate systems – and perhaps the number of cooling towers and circulating water systems – increases to match the number of gas turbines. For a multi-shaft combined cycle power plant there is only one steam turbine, condenser and the rest of the heat sink for up to three gas turbines; only their size increases. Having only one large steam turbine and heat sink results in low cost because of economies of scale. A larger steam turbine also allows the use of higher pressures and results in a more efficient steam

cycle. Thus the overall plant size and the associated number of gas turbines required have a major impact on whether a single shaft combined cycle power plant or a multiple shaft combined cycle power plant is more economical.

The combined-cycle system includes single-shaft and multi-shaft configurations. The single-shaft system consists of one gas turbine, one steam turbine, one generator and one Heat Recovery Steam Generator (HRSG), with the gas turbine and steam turbine coupled to the single generator in a tandem arrangement on a single shaft. Key advantages of the single-shaft arrangement are operating simplicity, smaller footprint, and lower startup cost. Single-shaft arrangements, however, will tend to have less flexibility and equivalent reliability than multi-shaft blocks. Additional operational flexibility is provided with a steam turbine which can be disconnected, using a synchro-self-shifting (SSS) Clutch, for start up or for simple cycle operation of the gas turbine.

Multi-shaft systems have one or more gas turbine-generators and HRSGs that supply steam through a common header to a separate single steam turbine-generator. In terms of overall investment a multi-shaft system is about 5% higher in costs.

Single- and multiple-pressure non-reheat steam cycles are applied to combined-cycle systems equipped with gas turbines having rating point exhaust gas temperatures of approximately 540 °C or less. Selection of a single- or multiple-pressure steam cycle for a specific application is determined by economic evaluation which considers plant installed cost, fuel cost and quality, plant duty cycle, and operating and maintenance cost.

Multiple-pressure reheat steam cycles are applied to combined-cycle systems with gas turbines having rating point exhaust gas temperatures of approximately 600 °C.

The most efficient power generation cycles are those with unfired HRSGs with modular pre-engineered components. These unfired steam cycles are also the lowest in cost.

Supplementary-fired combined-cycle systems are provided for specific application.

The primary regions of interest for cogeneration combined-cycle systems are those with unfired and supplementary fired steam cycles. These systems provide a wide range of thermal energy to electric power ratio and represent the range of thermal energy capability and power generation covered by the product line for thermal energy and power systems.

1.7 Efficiency of CCGT Plants

To avoid confusion, the efficiency of heat engines and power stations should be stated as Higher Heating Value (HHV) or Lower Heating Value (LHV) to exclude, or include respectively the heat and power that can be obtained from condensing the flue gas, and Gross output at the generator terminals or Net Output at the power station fence are being considered.

The LHV figure is NOT a computation of electricity net energy compared to energy content of fuel input; it is 11% higher than that. The HHV figure is a computation of electricity net energy compared to energy content of fuel input. If the LHV approach were used for some new condensing boilers, the efficiency would calculate to be over 100%. Manufacturers prefer to cite the higher LHV efficiency, e.g. 60%, for a new CCGT, but utilities, when calculated how much electricity the plant will generate, divide this by 1.11 to get the real, e.g. 54%, HHV efficiency of that CCGT. Coal plant efficiencies are computed on a HHV basis (it doesn't make nearly as much difference for coal burn, as for gas).

The difference between HHV and LHV for gas, can be estimated (using USA units) by $1055 \text{ Btu/Lb} \times w$, where w is the lbs of water after combustion per lb of fuel. To convert the HHV of natural gas, which is 23875 Btu/lb, to an LHV (methane is 25% hydrogen) would be: $23875 - (1055 \times 0.25 \times 18/2) = 21500$. Because the efficiency is determined by dividing the energy output by the input, and the input on an LHV basis is smaller than the HHV basis, the overall efficiency on an LHV basis is higher. So, using the ratio: $23875/21500 = 1.11$ you can convert the HHV to an LHV.

So a real best-of-class base load CCGT efficiency of 54%, as experienced by the utility operating the plant, translates to 60% LHV as the manufacturer's published headline CCGT efficiency. In general in service Combined Cycle efficiencies are over 50 percent on a lower heating value and Gross Output basis. Most combined cycle units, especially the larger units, have peak, steady state efficiencies on the LHV (marketing figure) basis of 55 to 59%. Research aimed at 1370 °C (2500 °F) turbine inlet temperature has led to even more efficient combined cycles and nearly 60 percent LHV efficiency (54% HHV efficiency) has been reached in the combined cycle unit of Baglan Bay, a GE H-technology gas turbine with a NEM 3 pressure reheat boiler, utilising steam from the HRSG to cool the turbine blades.

Siemens AG announced in May 2011 to have achieved a 60.75% net efficiency with a 578 megawatts SGT5-8000H gas turbine at the Irsching Power Station.

The most recent General Electric 9HA can attain 41.5% simple cycle efficiency and 61.4% in combined cycle mode, with a gas turbine output of 397 to 470MW and a combined output of 592MW to 701MW. Its firing temperature is between 2,600 and 2,900 °F (1,430 and 1,590 °C), its overall pressure ratio is 21.8 to 1 and is scheduled to be used by Électricité de France in Bouchain. The Chubu Electric's Nishi-ku, Nagoya power plant 405MW 7HA is expected to have 62% gross combined cycle efficiency.^[5]

By combining both gas and steam cycles, high input temperatures and low output temperatures can be achieved. The efficiency of the cycles add, because they are powered by the same fuel source. So, a combined cycle plant has a thermodynamic cycle that operates between the gas-turbine's high firing temperature and the waste heat temperature from the condensers of the steam cycle. This large range means that the Carnot efficiency of the cycle is high. The actual efficiency, while lower than this, is still higher than that of either plant on its own.^[6] The actual efficiency achievable is a complex area.^[7]

The electric efficiency of a combined cycle power station, if calculated as electric energy produced as a percent of the lower heating value of the fuel consumed, may be as high as 58 percent when operating new, i.e. unaged, and at continuous output which are ideal conditions. As with single cycle thermal units, combined cycle units may also deliver low temperature heat energy for industrial processes, district heating and other uses. This is called cogeneration and such power plants are often referred to as a Combined Heat and Power (CHP) plant.

1.7.1 Boosting efficiency

The efficiency of CCGT and GT can be boosted by pre-cooling combustion air. This is practiced in hot climates and also has the effect of increasing power output. This is achieved by evaporative cooling of water using a moist matrix placed in front of the turbine, or by using Ice storage air conditioning. The latter has the advantage of greater improvements due to the lower temperatures available.

Furthermore, ice storage can be used as a means of load control or load shifting since ice can be made during periods of low power demand and, potentially in the future the anticipated high availability of other resources such as renewable during certain periods.

1.8 Integrated gasification combined cycle (IGCC)

An integrated gasification combined cycle, or IGCC, is a power plant using synthesis gas (syngas). Syngas can be produced from a number of sources, including coal and biomass. The system utilizes gas and steam turbines, the steam turbine operating off of the heat leftover from the gas turbine. This process can raise electricity generation efficiency to around 50%.

1.9 Integrated solar combined cycle (ISCC)

An Integrated Solar Combined Cycle is a hybrid technology in which a solar field is integrated within a combined cycle plant. In ISCC plants, solar energy is used as an auxiliary heat supply, supporting the steam cycle, which results in increased generation capacity or a reduction of fossil fuel use.^[8]

Thermodynamic benefits are that daily steam turbine startup losses are eliminated.^[citation needed] Major factors limiting the load output of a combined cycle power plant are the allowed pressure and temperature transients of the steam turbine and the heat recovery steam generator waiting times to establish required steam chemistry conditions and warm-up times for the balance of plant and the main piping system. Those limitations also influence the fast start-up capability of the gas turbine by requiring waiting times. And waiting gas turbines consume gas. The solar component, if the plant is started after sunshine, or before, if we have heat storage, allows us to preheat the steam to the required conditions. That is, the plant is started faster and we consume less gas before achieving operating conditions. ^[9] Economic benefits are that the solar components costs are 25% to 75% those of a Solar Energy Generating Systems plant of the same collector surface.

1.10 Automotive use

Combined cycles have traditionally only been used in large power plants. BMW, however, has proposed that automobiles use exhaust heat to drive steam turbines.^[14] This can even be connected to the car or truck's cooling system to save space and weight, but also to provide a condenser in the same location as the radiator and preheating of the water using heat from the engine block.

It may be possible to use the pistons in a reciprocating engine for both combustion and steam expansion as in the Crower engine. A turbocharged car is also a combined cycle. Bowman of Southampton offer a commercially proven add-on turbocharger which additionally can generate electric power lowering overall fuel consumption by about 8%.

CHAPTER 2

2.0 Heat Recovery Steam Generators

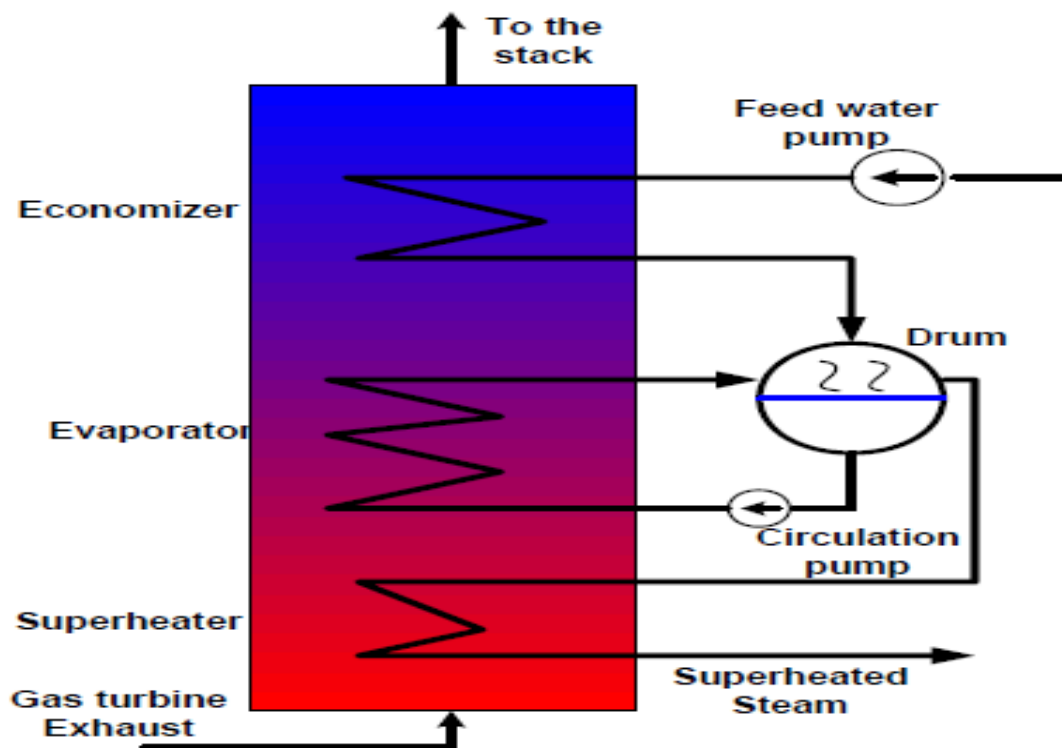
The link between the gas turbine and the steam cycle in a combined cycle is the Heat Recovery Steam Generator (HRSG). In the HRSG, heat from gas turbine exhaust is used to generate steam in the steam cycle. The HRSG is made up of several heat exchangers for feed water heating, water vaporization and steam superheating, while at the same time cooling the gas turbine exhaust. There is often more than one pressure level in the steam side of the HRSG.

2.1 Single Pressure HRSG

In a single-pressure HRSG, the heat recovery from gas turbine exhaust is normally done in three steps:

In the *economizer*, low value heat from the exhaust gas is used for heating the pressurized feed water;

In the *super heater*, high value heat from the exhaust is used for super heating the steam generated in the evaporator.



HRSG Flow Diagram

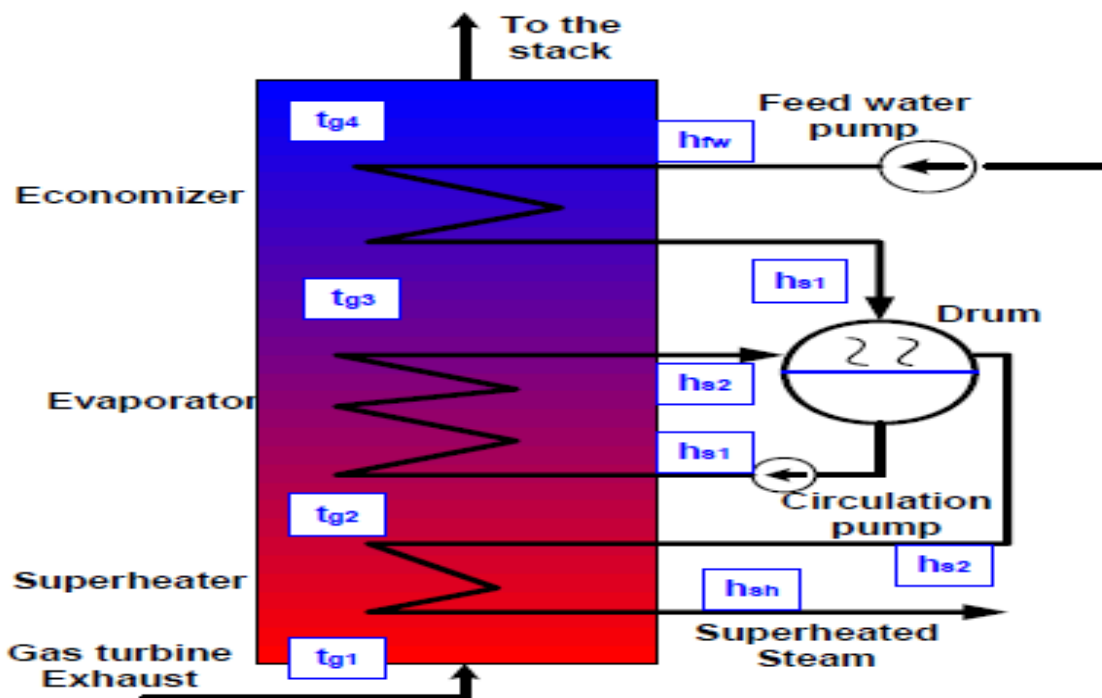
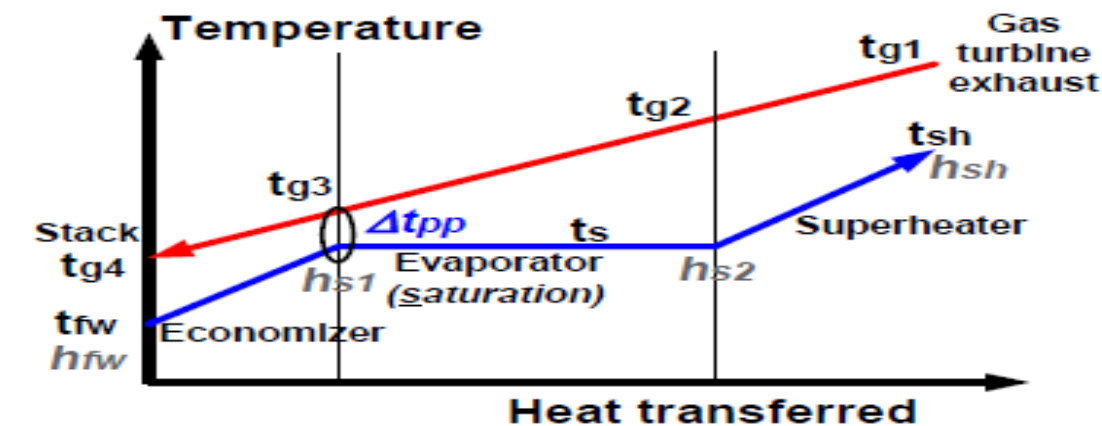
2.1.1 Heat Transfer Diagram and Heat Balances

The three steps of heat recovery in the HRSG can be illustrated in a temperature-heat transfer diagram. The aim of such a diagram is to facilitate analysis of different parameters in the HRSG. The upper line illustrates the cooling of exhaust gas from inlet to outlet in the HRSG.

The lower line illustrates the water-to- steam process, where evaporation takes place at constant temperature.

The area between the exhaust profile and the water profile is a measure of the heat loss in the HRSG. An important design parameter is the pinch point temperature difference, Δt_{pp} .

Different parameters of importance for analyzing the combined cycle are found through heat balances in the HRSG.



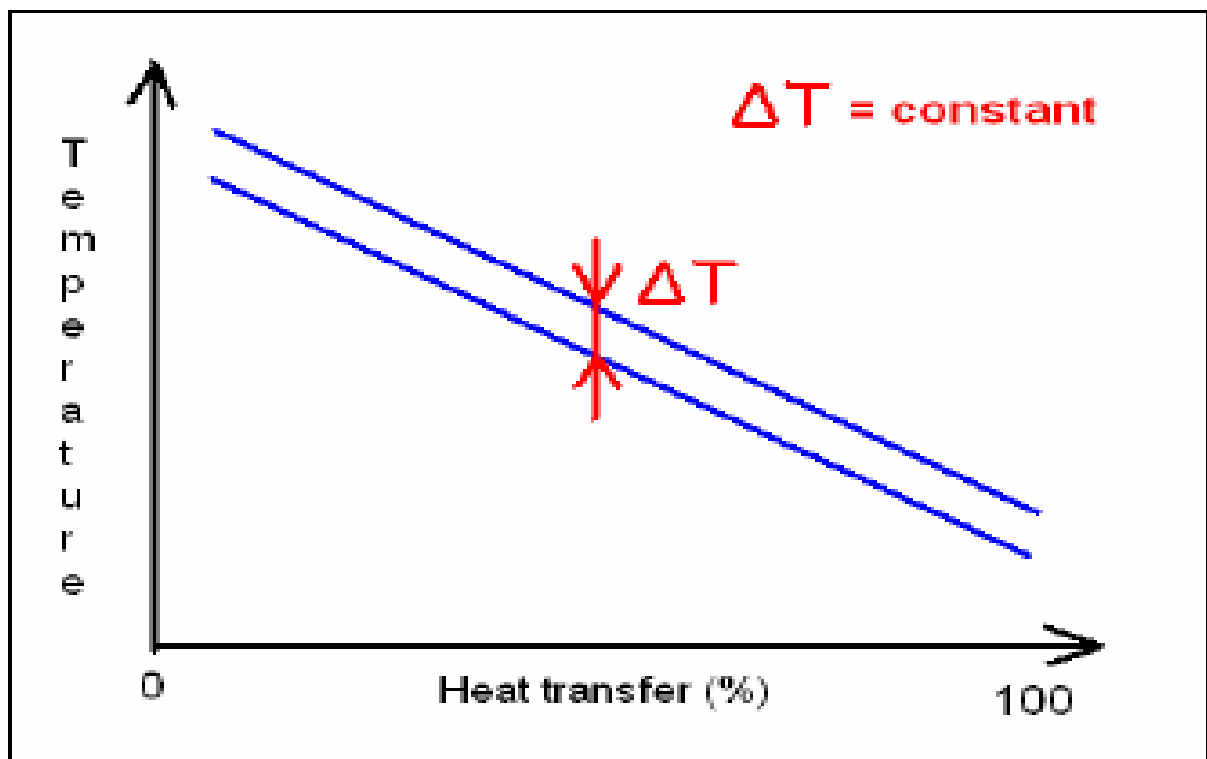
2.2 Measure of heat loss in the HRSG

In an *ideal heat exchanger*, the temperature difference between the two media on each heat exchanger side approaches zero, i.e. $\Delta T \rightarrow 0$

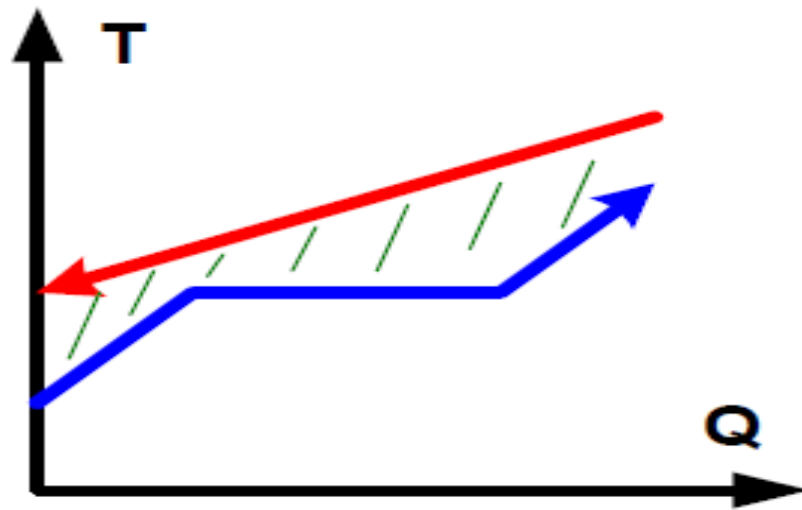
In a *real heat exchanger*, the temperature difference can never become zero, because the possible size of the heat exchanger surface area is limited.

The larger the temperature difference between the two media, the lower the efficiency of the heat exchange, and thus less heat is recovered. The lower the heat recovery (and thus efficiency), the higher the exit temperature of the cooled medium. In the HRSG of a combined cycle, the water vaporization at constant temperature constitutes the largest heat loss.

The larger the area between the exhaust profile and the water profile, the higher the exhaust temperature out from HRSG, i.e. the stack temperature.



IDEAL HEAT EXCHANGE



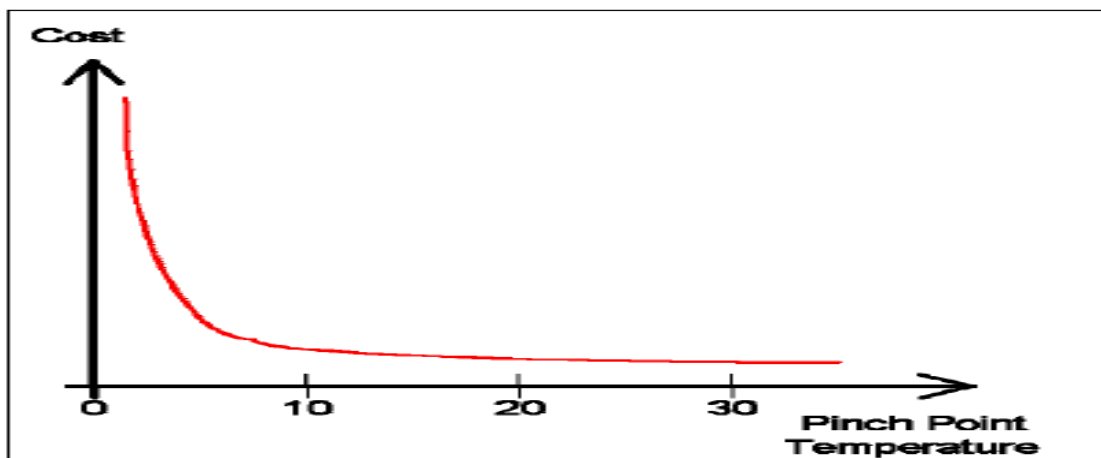
HEAT EXCHANGE IN A HRSG

2.2.1 Pinch point temperature difference, Δt_{PP}

The pinch point temperature difference (Δt_{pp}) is the difference between the temperature of the exhaust exiting the evaporator (t_{g3}) and the temperature of water evaporation (t_s):

$$\Delta t_{PP} = t_{g3} - t_s$$

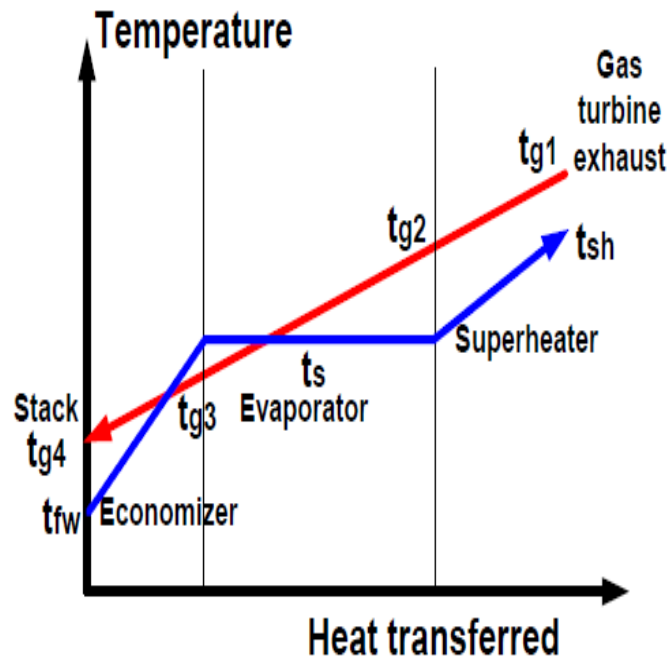
The water evaporation takes place at constant temperature (isothermal process) at the pressure prevailing in the drum. Without considering pressure losses in the water/steam tubes, the pressure in the boiler drum equals the pressure after the feed water pump. The pinch point temperature difference, Δt_{PP} , is usually between 5°C and 15°C. The lower the pinch point temperature difference, the higher the cost for the heat exchanger.



On the other hand, the lower the pinch point temperature difference, the more heat is recovered from the flue gas to the water-to-steam process. The aim in introducing the pinch point temperature difference into HRSG design is to avoid a temperature cross situation.

A temperature cross situation results when part of the economizer and part of the evaporator virtually HEAT the EXHAUST GAS, as illustrated below.

The temperature cross situation is thermodynamically impossible.



● Observe that all temperatures, tg_4 , t_{FW} , t_{g1} and t_{sh} , may be very reasonable.

The temperature cross situation can occur if the engineer assumes a stack temperature for his/her calculations without checking the pinch point temperature difference.

It is thus important for all HRSG designers to always base their calculations on the pinch point temperature difference and not on the stack temperature.

We can construct the composite curves by adding hot and cold curves moves horizontally towards others.

2.2.2 Approach temperature

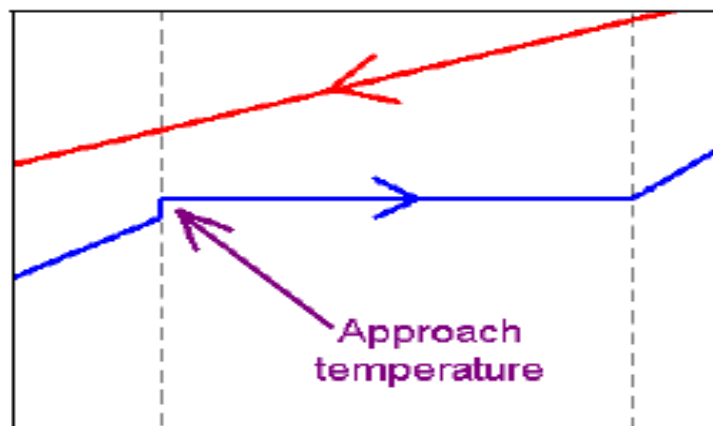
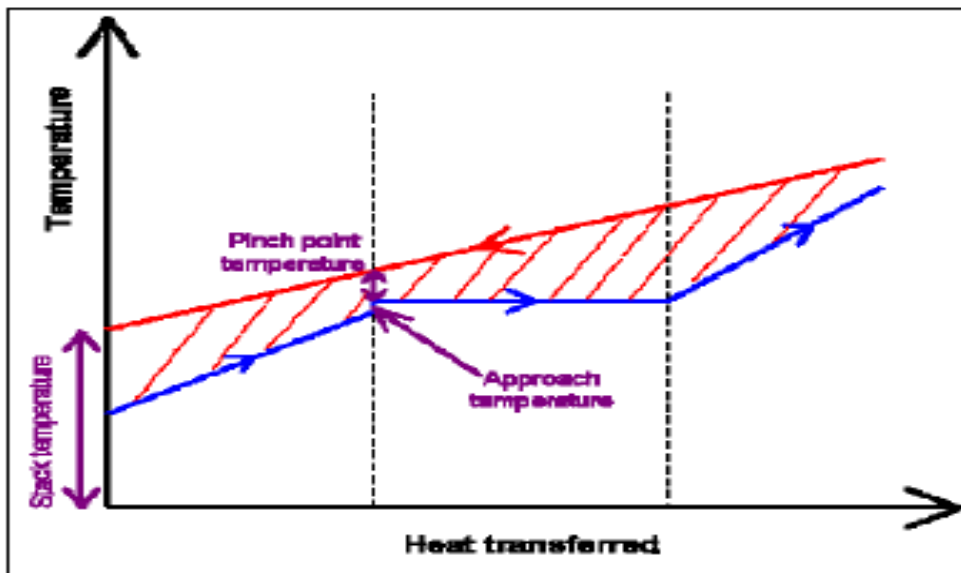
The approach temperature is defined as the temperature difference between the saturation temperature and the economizer outlet temperature, i.e.

$$\Delta t_{appr} = t_s - t_{eco,out}$$

The purpose of keeping the economizer outlet temperature somewhat lower than the saturation temperature is to avoid evaporation in the economizer.

Evaporation would cause tube erosion and other problems, since most economizers are not designed to handle two-phase flows (such a heat exchanger would cost significantly more!)

The approach temperature range is typically between 5°C and 12°C.



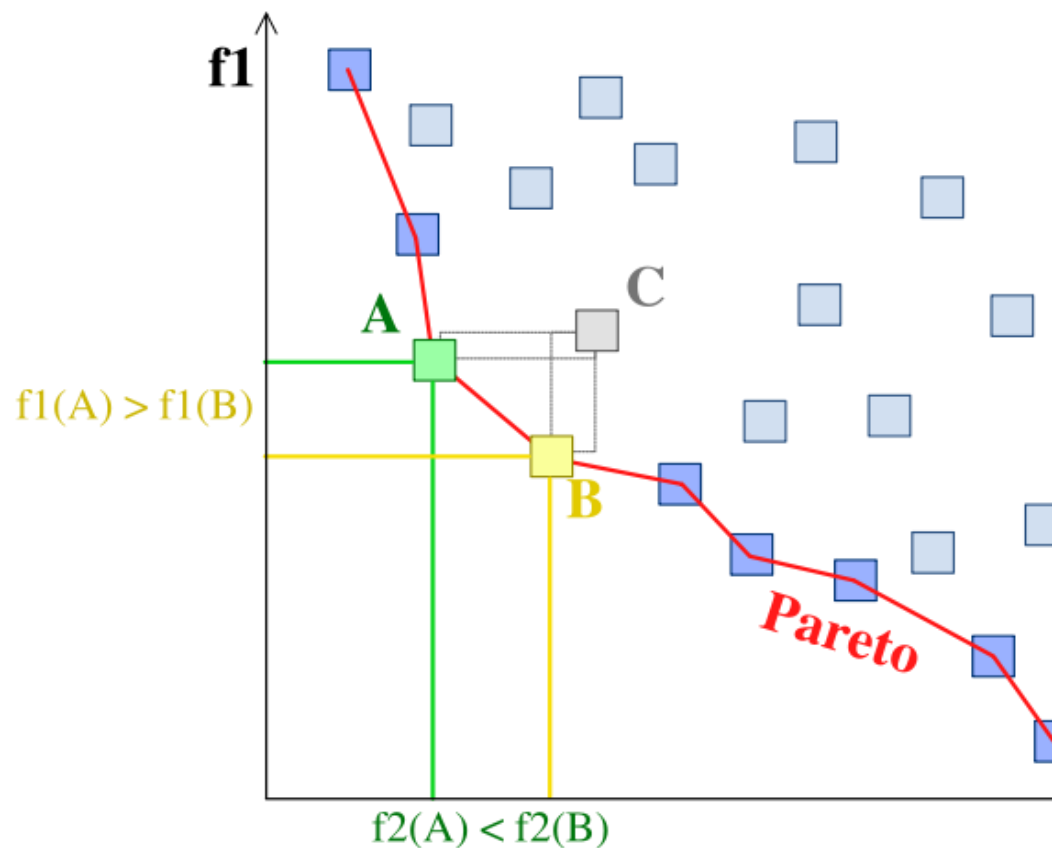
2.3 Heat balances in the HRSG

The heat recovered from the flue gas can be balanced both from the exhaust side view point and from the water profile view point according to:

$$Q_{\text{recovered}} \begin{cases} \nearrow m_{\text{gt}} \cdot c_{\text{pg}} \cdot (t_{\text{g1}} - t_{\text{g4}}) & \text{Gas side} \\ \searrow m_{\text{st}} \cdot (h_{\text{sh}} - h_{\text{fw}}) & \text{Steam side} \end{cases} \quad (\text{A})$$

Heat balances for each component in the HRSG are as follows (without the economizer approach temperature):

$$\bullet \text{ Economizer: } m_{\text{gt}} \cdot c_{\text{pg}} \cdot (t_{\text{g3}} - t_{\text{g4}}) = m_{\text{st}} \cdot (h_{\text{S1}} - h_{\text{fw}}) \quad (\text{B})$$



$$\bullet \text{ Evaporator: } m_{gt} \cdot c_{pg} \cdot (t_{g2} - t_{g3}) = m_{st} \cdot (h_{s2} - h_{s1}) \quad (C)$$

$$\bullet \text{ Superheater: } m_{gt} \cdot c_{pg} \cdot (t_{g1} - t_{g2}) = m_{st} \cdot (h_{sh} - h_{s2}) \quad (D)$$

Caution! The specific heat, c_p , is temperature dependent and thus taken as an average value between the two gas temperatures in each equation, i.e. c_p for the gas passing in the economizer is NOT the same as for the gas in the superheater (for example).

The enthalpies h_{s1} and h_{s2} are found in water/steam saturation tables for the pressure prevailing in the boiler drum. Without consideration of pressure losses, the drum pressure equals the steam turbine inlet pressure.

Heat balances for each component in the HRSG with the economizer approach temperature:

$$\bullet \text{ Economizer: } m_{gt} \cdot c_{pg} \cdot (t_{g3} - t_{g4}) = m_{st} \cdot (h_{appr} - h_{fw}) \quad (B)$$

$$\bullet \text{ Evaporator: } m_{gt} \cdot c_{pg} \cdot (t_{g2} - t_{g3}) = m_{st} \cdot (h_{s2} - h_{appr}) \quad (C)$$

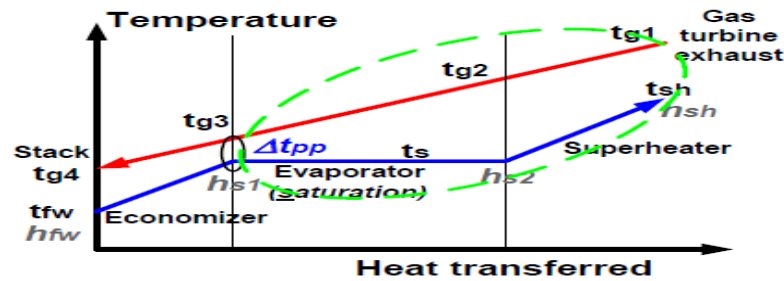
2.4 HEAT BALANCE FOR THE SUPERHEATER AND EVAPORATOR

For overview calculations of a combined cycle, the gas temperature, t_{g2} , is seldom needed. The pinch point temperature difference, on the other hand, is important to estimate, and if it is already defined, a heat balance may give other parameters of importance—for example, the steam flow.

Thus, by combining equations (C) and (D), a very convenient heat balance is obtained:

$$m_{gt} \cdot c_{pg} \cdot (t_{g1} - t_{g3}) = m_{st} \cdot (h_{sh} - h_{s1})$$

$$\text{where } \Delta t_{pp} = t_{g3} - t_s$$

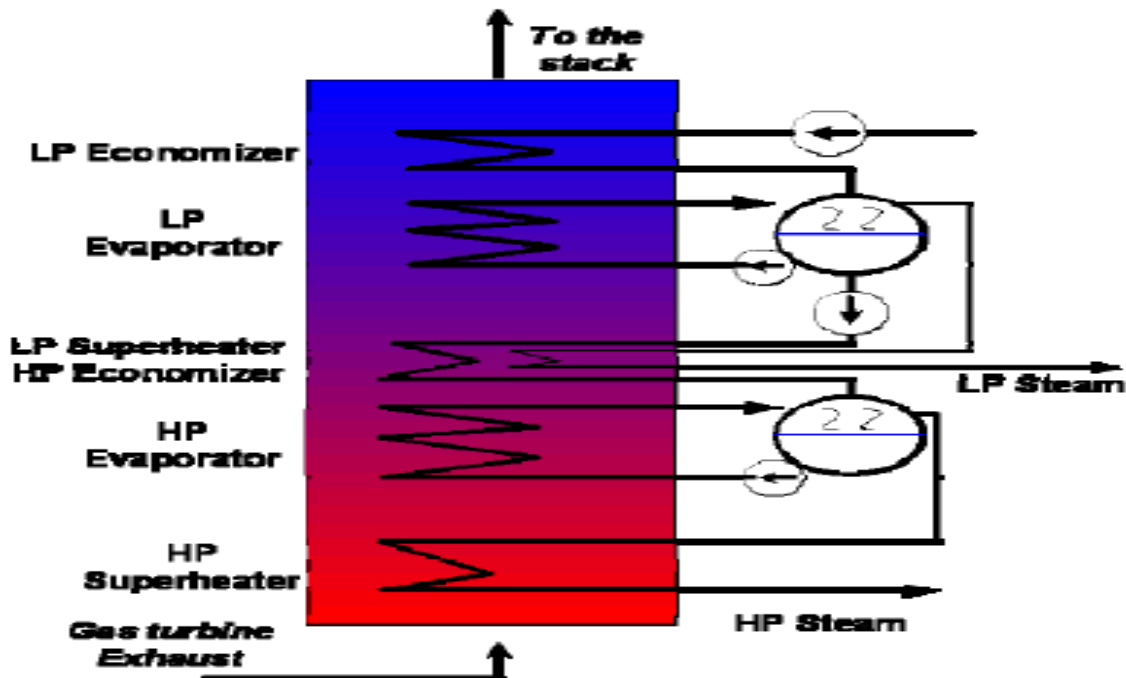


2.5 Multi-Pressure HRSG

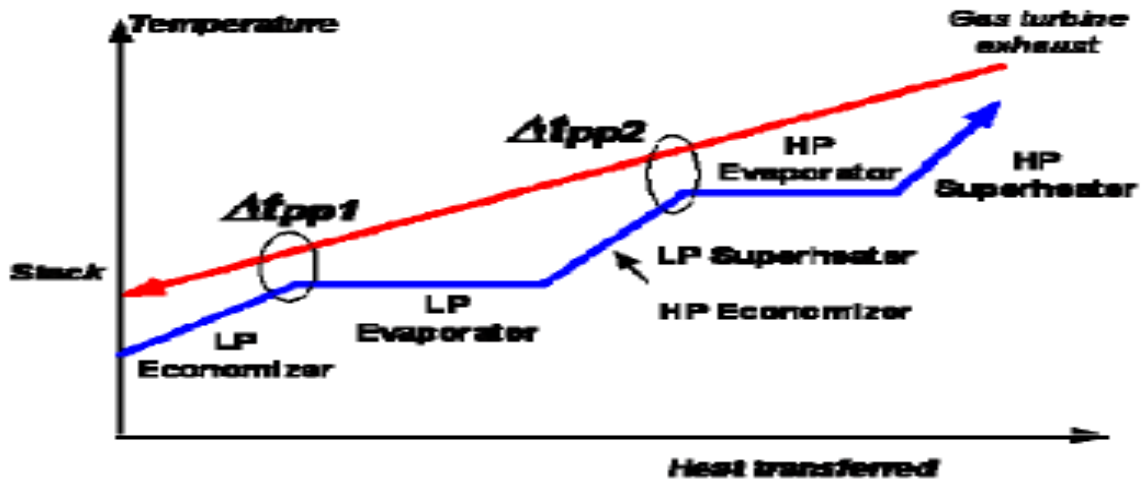
More heat from the exhaust gas can be recovered to the steam cycle, if several pressure levels in the steam cycle are introduced.

The increased heat exchange efficiency of a multi-pressure HRSG results in a lower stack temperature

Compared to a single pressure HRSG. For an HRSG with two pressure levels, the two pressures cannot be close in value; the pressure Difference should be distinctive.



HRSG with Two Pressure Levels

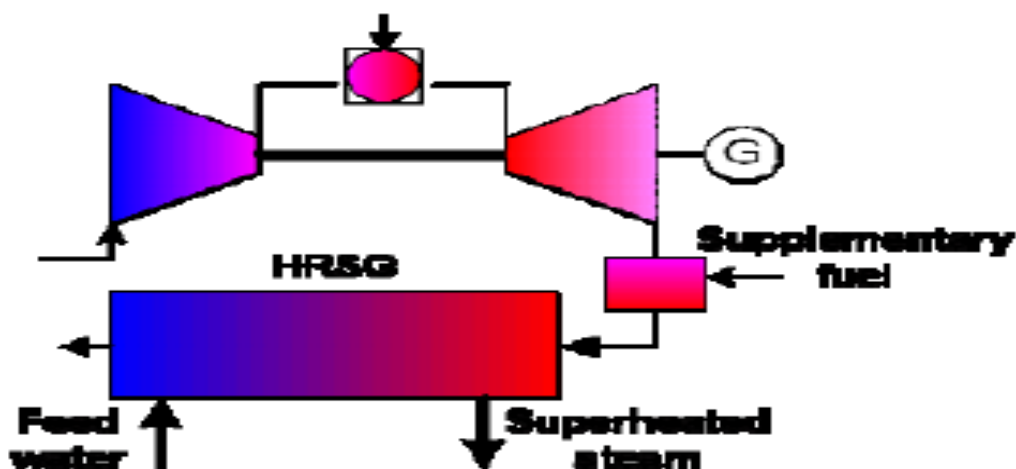


2.6 Supplementary Firing

Supplementary firing of gas turbine exhaust will increase the temperature, and thus more steam can be produced in the steam cycle. The supplementary fuel can be solid, liquid or gaseous and can be chosen independently of the gas turbine.

Supplementary firing comes with several advantages, particularly in a cogeneration plant.

Supplementary firing does not increase the combined cycle efficiency, since the fuel supplied is only used in the steam cycle part. This means that supplementary firing generally causes lower cycle efficiency.



2.7 Literature review

Energy is one of the main driving forces, which helps to sustain human life. Energy is available in several forms, e.g., heat, light, electricity. Energy, economic and environmental impact considerations have gained a lot of attentions during the last three decades. Energy resources in the market are getting less and at higher prices as the industrial revolution progress. This is due to several reasons such as the growth in global economy, depletion of energy resources, and the environmental impacts of energy production. Therefore, these energy issues are now threatening many aspects of human lives on the planet. Concerns exist regarding energy conversion from thermal sources to electrical sources. In this regard, power plants play main role in electricity production. Among different kinds of power plants, combined cycle power plants (CCPPs) gained a lot of attentions due to the fact that they are attractive in power generation field because of higher thermal efficiency rather than individual gas turbine or steam power plants. In addition, they are also important due to their relatively high energy efficiencies, low pollutant and greenhouse gas emissions, and operational flexibility. Literature on the subject shows that several efforts have been carried out on the plant optimizations. Energy consumption minimization, optimization of annual return and other approaches were used. Unlike energy, exergy is a measure of the quality of energy that can be considered to evaluate, analyze and optimize the system. Exergy analysis is utilized to define the maximum performance of a system and to specify its irreversibility. Therefore, exergy is a powerful tool for evaluating the performance of the cycle and also one objective approached usually with the lack of economic, technical and environmental feasibilities. Recently, combination of tools like exergy, economic and environmental assessments have received increasing attention around the world. Therefore the method which considered all the objectives and provides a feasible solution is currently needed by CCPPs plant designers. In this respect, different studies [1–8] have been carried out to investigate the combined cycle power plants, however a few multi-objective optimization of such plant which accounts for all the three main issues simultaneously have done to the best of our knowledge. Exergy analysis is a useful tool to find the locations, types and magnitudes of true inefficiencies (irreversibility's) and recommend ways to improve the overall efficiency of the system[2,9–12].In the literature, there have been various studies associated with

exergy analyses of plants. Lior proposed a concept about future power generation systems and the role of exergy analysis in their development. He illustrated some thoughts to meet the power demands under the constraints of increased population and land use when holding the environmental impact to a tolerable one. Next he focused on exergy analysis which would be essential in the conception and development of such processes. Finally he discussed about

Nomenclature

List of abbreviations

a	air	k	k th component
act	actual	LHV	lower heating value [kJ/kg]
c	cost per exergy unit [\$/MJ]	$\dot{m}$	mass flow rate [kg/s]
c_f	cost of fuel per energy unit [\$/MJ]	c	probability
$\dot{C}$	cost flow rate (\$/s)	Ph	physical
ch	chemical	PP	pinchpoint
c_p	specific heat at constant pressure [kJ/kg K]	Q	heat
CW	coolingwater	R	gas constant
D	destruction	ref	reference
Env	environmental	r_{comp}	compressor pressure ratio
$\dot{E}_x$	exergy flow rate [MW]	$\dot{W}_{net}$	net power output [MW]
$\dot{E}_{x_D}$	exergy destruction rate [MW]	Z	capital cost of a component [\$]
f	fuel	$\dot{Z}$	capital cost rate [\$/s]
F	fuel for a component	η_{comp}	compressor isentropic efficiency
g	gas	η_{cc}	combustion chamber first law efficiency
h	specific heat [kJ/kg]	η_{GT}	gas turbine isentropic efficiency
is	isentropic	C	specific heat ratio
j	j th stream	u	maintenance factor
		n	coefficient of Fuel Chemical exergy

which kind of development is essential for the power generation units in the coming century. The results indicated that exergy analysis will help the designers to come up with a good decision on how to improve the system performance. Rosen and Dincer conducted a comprehensive study of industrial steam heating process through exergy analysis. The results showed that using exergy analysis and the technical factors that influence the feasibility of steam supplied for other energy sources in industrial heating. They presented an example for the Bruce Energy Center in Ontario, Canada to demonstrate the importance of using exergy analysis to assess the feasibility of industrial steam process heating. The results suggested exergy analysis should be used as an important tool in process optimization while the use of large quantities of the steam in energy centers is considered. In recent decades, thermo economics and exergo economics have been increasingly utilized by researchers, combining thermodynamics with economics. Many such studies have been reported, especially for power generation.

Abusoglu and Kanoglu surveyed a review of exergo economic analysis and optimization of combined heat and power production. This paper was a review on the exergo economic analysis and optimization of combined heat and power production systems. They gather brief historical information on the exergo economics analysis and optimization. They also discussed the concept of exergy cost and cost accounting methods. Ahmadi and Dincer performed a comprehensive thermodynamic modeling of a dual pressure (HRSG). To find the best design parameters for HRSG, they applied a multi objective optimization. In this regard for this optimization two objective functions are considered. Results showed that increasing the drum pressures results to increase in HRSG exergy efficiency and a decrease in the HRSG exergy destruction rate. Nevertheless, increasing the pinch temperatures resulted in decrease in HRSG exergy efficiency and also increases in HRSG exergy destruction. One of the most important concerns for human is to reduce environmental impact of energy system and make use of sustain- able energy technologies to mitigate global warming. Some exergy methods are tried to reduce exergy losses and increase exergy efficiency, But there are some ways that exergy assist to understand and reduce environmental impact. In this regards, many reports in the literature consider environmental aspects of thermal systems . Lazzaretto and Toffolo optimized an energy system from energy, economy and environmental aspects. They defined an environmental impact objective function in terms of cost by weighting CO and NO emissions considering their unit damage costs. In optimization approach, three objective functions are selected to find the optimal solutions by using evolutionary algorithm. They applied an additional objective which was the cost related to NO_x and CO emissions in cost terms. In this sense, they proposed the non-abbreviated term thermo enviro economic for this objective. BarzegarAvval et al. studied the thermo economic environmental multi objective optimization of a gas turbine power plant using evolutionary algorithm. They used multi-objective optimization in order to optimize the system for the better performance assessment. Optimization results showed the overall exergo economic factor of the system increases from 32.79% to 62.24% compared with the actual power plant. In another research a comprehensive thermodynamic modeling and exergo environmental analysis of a tri generation system based on a micro an organic Rankine cycle and gas turbine for heating, cooling and electricity purposes are studied by Ahmadi et al.[27].

In this paper, mathematical model is developed for a combined cycle power plant. Energy, exergy, exergo economic and exergo environmental analyses of this power plant are conducted. For optimization process, three different objectives function namely exergy efficiency to be maximized, total cost rate of the system product and the CO₂ emission to be minimized are defined and optimized. The optimization is applied for three cases with different qualities at steam turbine outlet. Since the emission and efficiency have an inverse correlation and can be expressed as a function of each other, the Pareto frontier of these three cases is presented as a 3D line instead of a surface. Thereafter, energy efficiency, exergy efficiency, main work flows, purchase cost of each components and emission of these three different cases are conducted and compared. Therefore, the main objective of this research paper is to carry out the thermodynamic modeling and effects of steam turbine out- let quality on the output power of a combined cycle power plant. The sub objectives of this research study follow:

- A development model for a CCGT based on thermodynamic principles and energy balances.
- Considering three different objective functions to handle the optimization process for three different cases with different steam quality at steam turbine outlet.
- To have a good verification results, energy efficiency, exergy efficiency of the mentioned cases are evaluated and compared.
- The environmental impacts for three different cases with different steam quality at steam turbine outlet are determined.

2.8 Description of case study

Fig.1 illustrates a CCGT containing an air compressor (AC), combustion chamber (CC), a gas turbine (GT), a duct burner (DB), a double pressure heat recovery steam generator (HRSG) to produce superheated steam, a steam turbine (ST), a condenser (Cond) and pump (P). The output power of this power plant is about 415.1 MW.

CHAPTER - 3

3. Modeling and analysis

The comprehensive thermodynamic modeling, exergy analysis, and environmental impact assessments of this CCPP provided useful insights and it's essential for our optimization.

3.1. Energy analysis

Energy analysis is used to determine the temperature profile in the plant. We need the properties of each point of the cycle to do the exergy analysis. Exergy analysis is used to find the exergy efficiency and exergy destruction rate and also it is necessary to do the exergo economic and environmental analysis. Energy analysis is done usually through the physical modeling. The major portion of this modeling consists of the application of the mass and energy balance equations. The energy balance equations for various parts of the CCPP (Fig. 1) are as described in the following subsections.

3.1.1. Air compressor (AC)

Air enters to the air compressor at ambient temperature and pressure. The outlet temperature is a function of compressor inlet temperature T_1 , compressor isentropic efficiency η_{AC} , compressor pressure ratio r_C and specific heat ratio γ_a .

$$T_2 = T_1 \left\{ 1 + \frac{1}{\eta_{AC}} \left[r_C^{\frac{\gamma_a-1}{\gamma_a}} - 1 \right] \right\} \quad \dot{W}_{AC} = \dot{m}_a c_{pa} (T_2 - T_1) \quad (1)$$

3.1.2. Combustion chamber

The outlet properties of the combustion chamber are a function of air mass flow rate, fuel lower heating value (LHV) and combustion efficiency (η_{CC}), and are related as follows:

$$\dot{m}_a h_3 + \dot{m}_{f-CC} \text{LHV} = \dot{m}_g h_4 + (1 - \eta_{CC}) \dot{m}_{f-CC} \text{LHV} \quad (2)$$

3.1.3. Gas turbine

The gas turbine outlet temperature can be written as a function of gas turbine isentropic efficiency (η_{GT}), the gas turbine inlet temperature (GTIT) and gas turbine pressure ratio (P_3/P_4) as follows:

$$\dot{W}_{GT} = \dot{m}_g \cdot c_{pg}(T_3 - T_4) \quad \frac{P_4}{P_3} = (1 - \Delta p_{cc}) \quad (3)$$

3.1.4. Duct burner(DB)

For this CCPP the hot gases, before entering the heat recovery steam generator, enter the duct burner to raise their temperature around 70 °C. The outlet properties of the duct burner are a function of gas mass flow rate, fuel lower heating value (LHV) and duct burner efficiency (η_{DB}), and are related as follows:

$$\dot{m}_g h_4 + \dot{m}_{f-DB} \text{LHV} = (\dot{m}_g + \dot{m}_{f-DB}) h_5 + (1 - \eta_{DB}) \dot{m}_{f-DB} \text{LHV} \quad (4)$$

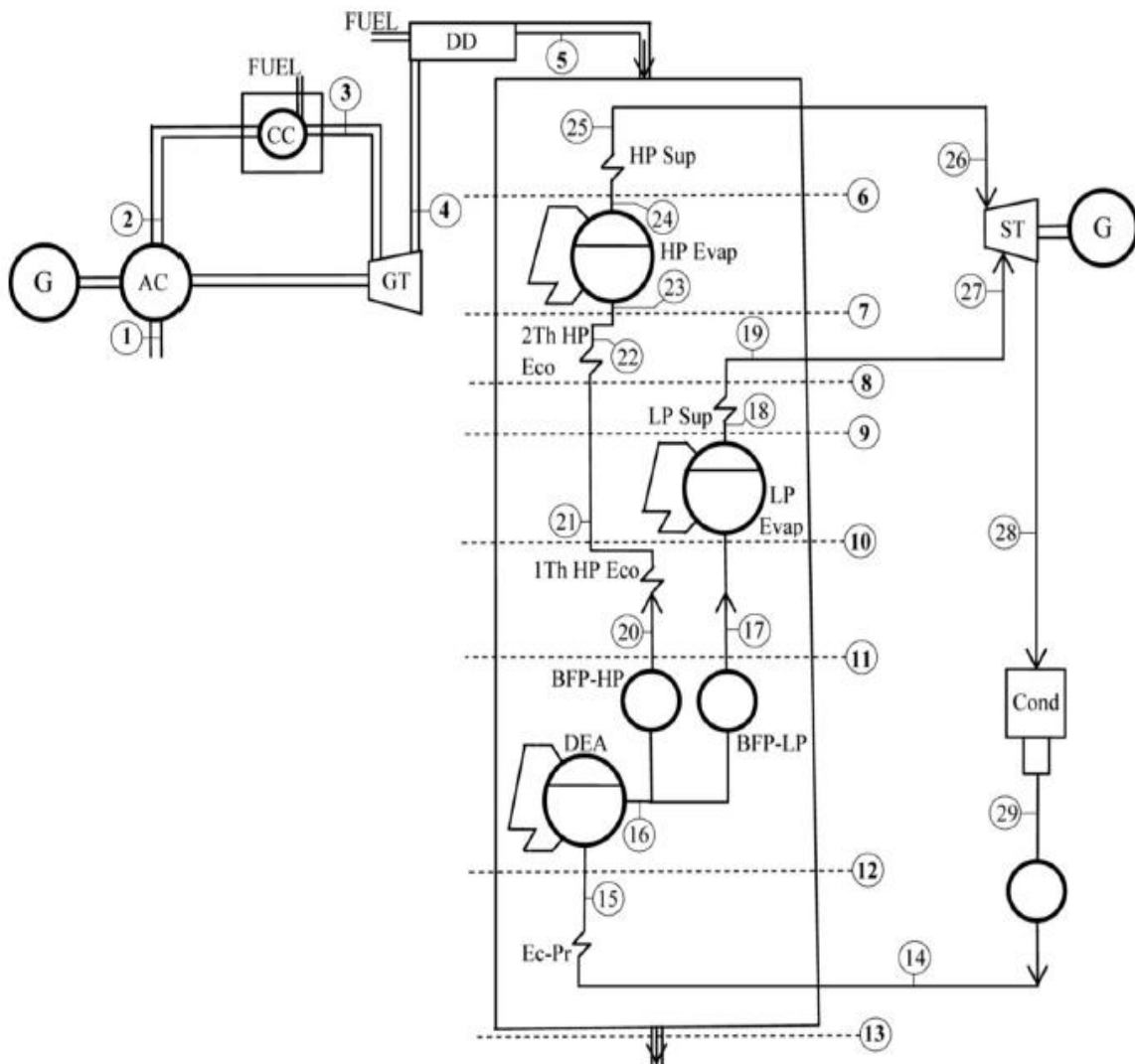


Fig. 1. Thermal schematic diagram of Neka combined cycle with dual pressure HRSG and supplementary firing unit.

the difference between the temperature of the gas at the entrance of the evaporator (economizer side) and the saturation temperature. The dual-pressure HRSG has two pinch points (PPHP and PPLP). The temperature differences between the water leaving the economizers and the saturation temperature are the approach points (APHP and APLP), which depend on the economizer's tube layout. Note that the pinch point and approach temperatures are considered constant here. Energy balances for each element of the HRSG are expressed as follows:

$$\dot{m}_g c_{pg}(T_7 - T_8) = \dot{m}_{s,HP}(h_{22} - h_{21}) \quad (7)$$

- Low-Pressure super heater (LP SH)

$$\dot{m}_g c_{pg}(T_8 - T_9) = \dot{m}_{s,LP}(h_{19} - h_{18}) \quad (8)$$

- Low-Pressure evaporator (LP Evap)

$$T_{10} = T_{18} + LP_{pinchpoint} \quad T_{17} = T_{18} - LP_{Approachpoint}$$

$$\dot{m}_g c_p(T_9 - T_{10}) = \dot{m}_{s,LP}(h_{18} - h_{17}) \quad (9)$$

- First High-Pressure economizer (1st HP Eco)

$$\dot{m}_g c_p(T_{10} - T_{11}) = \dot{m}_{s,HP}(h_{21} - h_{20}) \quad (10)$$

- Deaerator (DEA)

$$T_{15} = T_{16} + DEA_{Approachpoint} \quad \dot{m}_g c_p(T_{11} - T_{12}) = \dot{m}_s(h_{16} - h_{15}) \quad (11)$$

- Economizer Preheated (Ec-Pr)

$$\dot{m}_g c_p(T_{12} - T_{13}) = \dot{m}_s(h_{15} - h_{14}) \quad (12)$$

3.1.6. Steam turbine

Energy balance of the steam turbine is expressed as follows

$$\dot{m}_{26}h_{26} + \dot{m}_{27}h_{27} = \dot{W}_{ST} + \dot{m}_{28}h_{28} \quad \eta_{ST} = \frac{\dot{W}_{ST,act}}{\dot{W}_{ST,is}} \quad (13)$$

3.1.7. Condenser

Energy balance of the condenser is expressed as follows:

$$\dot{m}_w h_{28} + \dot{m}_{CW} h_{30} = \dot{m}_w h_{29} + \dot{m}_{CW} h_{31} \quad \eta_{\text{pump}} = \frac{h_{14(s)} - h_{29}}{h_{14} - h_{29}} \quad (14)$$

3.1.8. Pump

Energy balance of the pump is expressed as follows:

$$\dot{m}_w h_{29} = \dot{W}_p + \dot{m}_w h_{28} \quad (15)$$

3.2. Exergy analysis

Unlike energy, exergy is a measure of the quality of energy that can be considered to evaluate, analyze and optimize the system. Exergy analysis also can help develop strategies and guidelines for more efficient and effective use of energy, and is utilized to study various thermal processes, especially power generation. Applying the first and second laws of thermodynamics, the following exergy balance can be obtained.

$$\dot{E}x_Q = \sum_i \dot{m}_i ex_i = \sum_e \dot{m}_e ex_e + \dot{E}x_W + \dot{E}x_D \quad (16)$$

where the subscripts e and i denote the inlet and outlet of a control volume $\dot{E}x_D$ is the exergy destruction and other terms are as follows [25,28]:

$$\dot{E}x_Q = \left(1 - \frac{T_o}{T_i}\right) \dot{Q}_i \quad (17)$$

$$\dot{E}x_W = \dot{W} \quad (18)$$

$$ex = ex_{ph} + ex_{ch} \quad (19)$$

$$ex_{ph} = (h - h_o) - T_o(s - s_o) \quad (20)$$

$$ex_{mix}^{ch} = \sum_{i=1}^n X_i ex_i^{ch} + RT_o \sum_{i=1}^n X_i \ln X_i \quad (21)$$

Here, $\dot{Ex}_Q$ and $\dot{Ex}_W$ are the corresponding exergy rates associated with heat transfer and work across the boundary of a control volume respectively and T is the absolute temperature and the subscript 0 refers to the reference environment conditions. The reference environment considered here is $T_0 = 20^\circ\text{C}$ and $P_0 = 1.01\text{ bar}$.

To evaluate fuel exergy the above equation cannot be used. Therefore, the fuel exergy is extracted with the following [29]:

$$\xi = \frac{ex_f}{LHV_f} \quad (22)$$

The ratio of chemical exergy to LHV_f is usually close to unity for common gaseous fuels [29]:

$$\xi_{\text{CH}_4} = 1.06 \quad \xi_{\text{H}_2} = 0.985 \quad (23)$$

For a general gaseous fuel with composition C_xH_y , the following experimental correlation can be used for ξ [30].

$$\xi = 1.033 + 0.0169y/x - 0.0169/x \quad (24)$$

3.3. Exergoeconomic analysis

The main concept of thermoeconomics is the exergetic cost and it deals with cost accounting methods [22,31,32]. Therefore, a parameter called flow cost rate $\dot{C}$ is defined for each flow in the system, and a cost balance is written for each component.

$$\dot{C}_{q,k} + \sum_i \dot{C}_{i,k} + \dot{Z}_k = \sum_e \dot{C}_{e,k} + \dot{C}_{w,k} \quad \text{Where} \quad \dot{C}_j = c_j \dot{Ex}_j \quad \text{and} \quad \dot{Z}_k = \frac{Z_k \cdot \text{CRF} \cdot \phi}{N \times 3600} \quad (25)$$

where the purchase cost of the k th component is Z_k , the annual number of operation hours for the unit is N , the maintenance factor is ϕ which is often 1.06 and CRF is the capital recovery factor. Capital recovery factor depends on equipment life time and the interest rate which is determined here as follows [25]:

$$\text{CRF} = \frac{i \times (1 + i)^n}{(1 + i)^n - 1}$$

Where i and n are the interest rate and the total operating period of the system in years respectively. By using cost balance equations for each component separately, a set of linear equations are produced which their simultaneous solution results in streams cost.

3.4 Exergo environmental analysis

An important measure for reducing environmental impact, including emissions of carbon dioxide, a primary greenhouse gas, is increasing efficiency and thereby decreasing fuel use. Based on literature many studies are involved exergy and exergo economic analyses for CCPP, but many do not consider the environmental impact in their studies. Addressing this deficiency is one objective of this article, in which emissions of CO₂, CO and NO_x are considered. NO_x formation and CO emission are more accurately related to the primary combustion parameters. Therefore, adiabatic flame temperature and also resident time in this region affect the NO_x and CO emission. High resident time increases NO_x but reduces the CO value. Thus, the analytical prediction of gas turbine combustion emission is provided. In this regard, the first step is definition on important parameters like adiabatic flame temperature. The adiabatic flame temperature in the initial zone of the combustion chamber is determined as follows :

$$T_{pz} = A\sigma^\alpha \exp(\beta(\sigma + \lambda)^2) \pi^{x^*} \theta^{y^*} \psi^{z^*} \quad (27)$$

where π is dimensionless pressure (P/P_{ref}), θ is dimensionless temperature (T/T_{ref}), and ψ is the H/C atomic ratio, $\sigma = \phi$ for $\phi \leq 1$ (ϕ is mass or molar ratio) and $\sigma = \phi - 0.7$ for $\phi \geq 1$. Moreover, x , y and z are quadratic functions of σ based on the following equations:

$$x^* = a_1 + b_1\sigma + c_1\sigma^2 \quad (28)$$

$$y^* = a_2 + b_2\sigma + c_2\sigma^2 \quad (29)$$

$$z^* = a_3 + b_3\sigma + c_3\sigma^2 \quad (30)$$

where A , α , β , λ , a_i , b_i and c_i are constants. Values for these parameters are listed in Table 1.

The amount of CO and NO_x produced in a combustion chamber and combustion reaction is dependent on the adiabatic flame temperature. The pollutant emissions (in grams per kilogram of fuel) can be identified as follows [34]:

$$\begin{aligned} m_{NO_x} &= \frac{0.15 \times 10^{16} \tau^{0.5} \exp(-71100/T_{pz})}{P_3^{0.05} (\Delta P_3/P_3)} \\ m_{CO} &= \frac{0.179 \times 10^9 \exp(7800/T_{pz})}{P_3^2 \tau (\Delta P_3/P_3)} \end{aligned} \quad (31)$$

where s is the residence time in the combustion zone (assumed constant and equal to 0.002 s) [2], T_{pz} is the primary zone at combustion, P_3 is the combustor inlet pressure, and $DP_3 = P_3$ is the non-dimensional pressure drop in the combustion chamber.

To calculate the value of CO₂ emission, it is better to define the net emission per kW h produced energy of the plant and try to reduce it. This definition is a measure of environmental efficiency of the plant. This value talks about, how efficient the power plant is in terms of green house gases (GHG) emission. The value of CO₂ emission in combustion chamber can be calculated by using the following simple equation:

$$\dot{m}_{CO} = 44.01x \left(\frac{\dot{m}_{fuel}}{m_{fuel}} \right) \quad (32)$$

where x is the mole fraction of carbon in the fuel and m_{fuel} is the molar weight of the fuel. This simple equation estimated the CO₂

Table 1
Values for parameters in Eqs. (27)–(30).

Parameter	$0.3 \leq \varphi \leq 1.0$		$1.0 \leq \varphi \leq 1.6$	
	$0.92 \leq \theta \leq 2$	$2 \leq \theta \leq 3.2$	$0.92 \leq \theta \leq 2$	$2 \leq \theta \leq 3.2$
A	2361.76	2315.75	916.82	1246.17
α	0.1157	−0.049	0.288	0.381
β	−0.948	−1.114	0.145	0.347
λ	−1.097	−1.180	−3.277	−2.0365
a_1	0.014	0.010	0.031	0.036
b_1	−0.055	−0.045	−0.078	−0.085
c_1	0.052	0.0482	0.0497	0.0517
a_2	0.395	0.568	0.025	0.009
b_2	−0.441	−0.550	0.260	0.502
c_2	0.141	0.132	−0.131	−0.247
a_3	0.005	0.011	0.004	0.017
b_3	−0.129	−0.129	−0.178	−0.189
c_3	0.082	0.084	0.098	0.1037

Emission accurately for complete combustion. In this case the total carbon dioxide is the sum of carbon dioxide produced in duct burner and gas turbine combustion.

CHAPTER - 4

4. Optimization

A multi-objective optimization method is applied to find the best design parameters of this CCPP. Objective functions, decision variables and constraints of this case are represented in this section

4.1.2. Total cost rate

By multiplying the respective flow rates of CO, NO_x and CO₂ emission in their corresponding unit damage cost (Table 2), the environmental impact is known as the total pollution damage (\$/s) in the second objective function. Therefore, in this research the cost of pollution damage is directly added to the expenditures that must be paid.

$$\dot{C}_{Total} = \dot{C}_F + \sum_k \dot{Z}_k + \dot{C}_{Env} + \dot{C}_D \quad (34)$$

where

$$\dot{C}_{env} = c_{CO} \dot{m}_{CO} + c_{NO_x} \dot{m}_{NO_x} + c_{CO_2} \dot{m}_{CO_2} \quad (35)$$

Table 2
Environmental damage costs [28].

Emissions	CO	NO _x	CO ₂
Cost (\$/Kg)	6.242	8.175	0.024

$$\dot{C}_f = c_f \dot{m}_f \times \text{LHV} \quad (36)$$

$$\dot{C}_D = c_F \dot{E}x_D \quad (37)$$

In which $\dot{Z}_k$ is the purchase cost of each component and $\dot{C}_f$, $\dot{C}_D$ are fuel cost and cost of exergy destruction, respectively.

4.1.3. CO₂ emission

Since the importance of environmental effects, CO₂ emission is considered as the third objective function which is produced in the combustion chamber and duct burner. Therefore, the CO₂ emission of the plant can write as follow.

$$\varepsilon = \frac{\dot{m}_{CO_2}}{\dot{W}_{net}} \quad (38)$$

4.2. Physical constraints

There are some physical constraints that must also be taken into account in cycle design. For instance, condensate pump pressure outlet must be less than feed pump first stage pressure outlet. And also temperature at heat exchangers in HRSG must follow the second law analysis or cold line temperature cannot be greater than hot line temperature in steam cycle.

4.3. Decision variables

The decision variables are selected as 15 variables which define the thermodynamic of the cycle. Although the decision variables may be varied in the optimization procedure, each is normally required to be within a reasonable range. Such constraints, for these decision variables are listed in [Table 4](#).

4.4. Evolutionary algorithm: genetic algorithm

There are lots of optimization techniques, among them evolutionary algorithm (EA) is known as one the important optimization techniques. They are a set of modern metaheuristics used successfully in many applications with great complexity. Therefore, EAs are successful on solving difficult problems [\[35,36\]](#).

Multi objective Optimization with Two Objectives

This example has a two-objective fitness function $f(x)$, where x is also two-dimensional:

function $f = \text{mymulti1}(x)$

$$f(1) = x(1)^4 - 10 \cdot x(1)^2 + x(1) \cdot x(2) + x(2)^4 - (x(1)^2) \cdot (x(2)^2);$$

$$f(2) = x(2)^4 - (x(1)^2) \cdot (x(2)^2) + x(1)^4 + x(1) \cdot x(2);$$

Create this function file before proceeding, and store it as mymulti1.m on your MATLAB® path.

CHAPTER – 5

5. Results and discussion

5.1 Optimization result and the effect of steam turbine outlet vapor Fraction

Each optimization process has some constraints and the constraints in this case are the upper and lower bounds along with several physical criteria. Besides these constraints, the steam

Table 3
The list of constraints for optimization [4].

Constraints	Rationale
$GTIT < 1500 \text{ K}$	Material Temperature Limit
$r_{Comp} < 0.9$	Commercial Availability
$\eta_{Comp} < 0.9$	Commercial Availability
$\eta_{GT} < 0.9$	Commercial Availability
$P_{main} < 110 \text{ bar}$	Commercial Availability
$0.75 < \eta_{ST} < 0.9$	Commercial Availability
$0.75 < \eta_{Pump} < 0.9$	Commercial Availability
$m_{DB} < 2 \text{ kg/s}$	Super Heater Temperature Limitation
$5 \text{ bar} < P_{Cond} < 15 \text{ bar}$	Thermal Efficiency Limitation
$T_{main} < 550 \text{ °C}$	Material Temperature Limitation
$120 \text{ °C} \leq T_{13}$	To avoid formation of sulfuric acid in exhaust gases
$5 \text{ °C} < PP < 30 \text{ °C}$	Second Law of Thermodynamic Limitation

Table 4
List of decision variables with the upper and lower bounds [4].

Decision variables	Lower bound	Upper bound
Compressor pressure ratio	3	20
Compressor isentropic efficiency	0.75	0.94
Combustor outlet temperature (K)	1000	1500
Turbine efficiency	0.75	0.94
Duct burner fuel flow (kg/s)	0	1
HP Pinch point	1	8
LP Pinch point	10	32
Lower pressure steam mass flow (kg/s)	5	15
High pressure steam mass flow (kg/s)	50	100
Feed pump first stage pressure (bar)	5	15
Feed pump second stage pressure (bar)	90	120
Condensate pump pressure (bar)	2	8
Pump efficiency	0.6	0.9
Steam turbine efficiency	0.75	0.94
Condenser pressure (bar)	0.05	0.15

turbine outlet vapor quality is another really important parameter which greatly affects the optimization results. In the current design point, quality of the vapor at the turbine outlet is about 88%. It is really important to keep the quality of the vapor constant for the results to be more realistic and also optimization and data are more technically feasible and applicable.

Three cases with different qualities are considered in order to enhance the understanding on the effect of quality on the optimization results. For the first case, no constraint on the vapor quality at steam turbine outlet is considered and for the second case 80% quality at turbine outlet and for the last case which is the most realistic one, the quality of 88% is considered. Figs. 2–4 show the Pareto frontier for the three mentioned cases. The results of the vapor quality of 80% and the one with no constraints at turbine outlet are shown to be close together. In these three cases, the Pareto frontier is a 3D line not a surface. This is an expected result since the emission and efficiency have an inverse correlation and can be expressed as a function of each other. So, the dimension of the Pareto frontier which is supposed to be a surface with 2 dimensions is reduced due to this correlation. Furthermore, because of a closely linear relationship between emission and efficiency, Pareto curve relay on a plan in a 3D space. This is the plan of possible solutions for the given problem and constraints in terms of cost, emission and efficiency. Any point out of this plan is not a case of feasible plant with specific data and constraints.

For each case a fitted surface is used to represent the feasible solutions. The fitted parameters are listed in Table 5. In this table, an increase in the first constant indicate that an increase in the outlet vapor quality. Therefore, the optimum results reach a higher

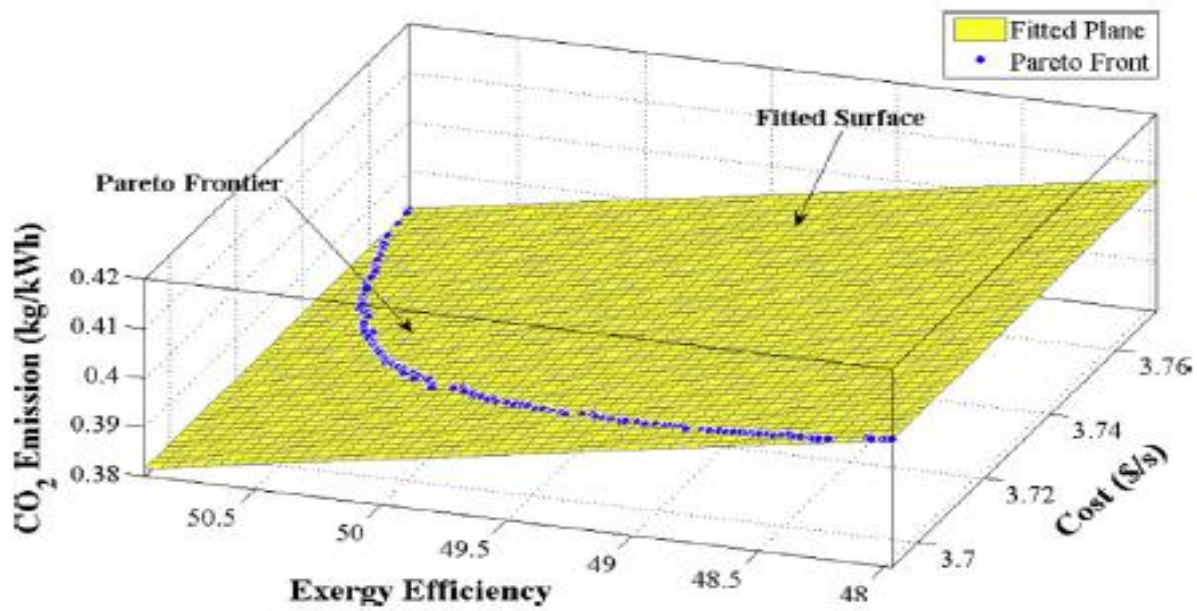
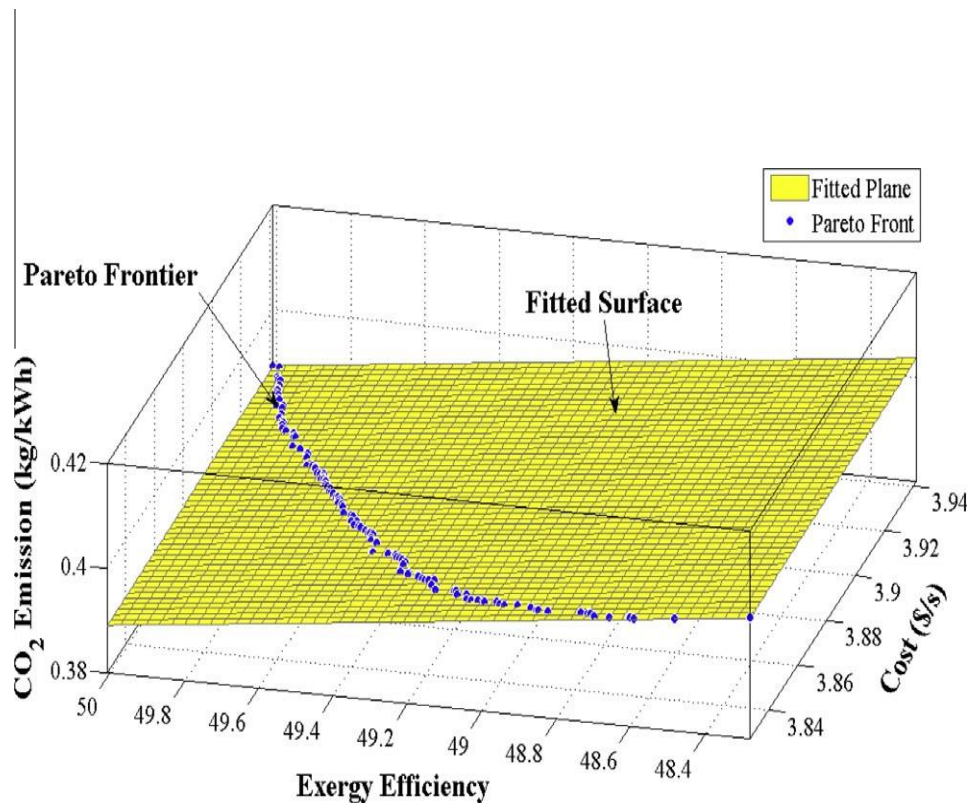


Fig. 2. 3D frontier and plan of feasible solutions for the case of no constraint on the quality.



5.2 Pareto frontier and selection of best point

Pareto frontier contains best set of solution family of the problem. Selecting the best point is not a clear and a general approach as it is dependent on the procedure of decision making and the changing of parameters. However a general approach is using the concept of equilibrium point. Equilibrium point is the interaction of the asymptotic surface of the Pareto frontier. A solution which is closer to this point is referred to be the best solution. Since the data lies on a 3D plan, one way to reduce the complexity of finding the equilibrium point is by transforming the coordinates and the Pareto curve to the corresponding fitted plan coordinates. Through this method, the problem changes to a 2D problem and after the optimum point is found, data are transformed back to the original 3D plan.

As discussed, the data from the 3D space was transformed into the 2D plan coordinates and the third coordinate was eliminated. The modified cost and efficiency values were plotted and the equilibrium and the best point solutions are calculated and selected. Figs. 5–7 show the procedure of selecting the best solution for

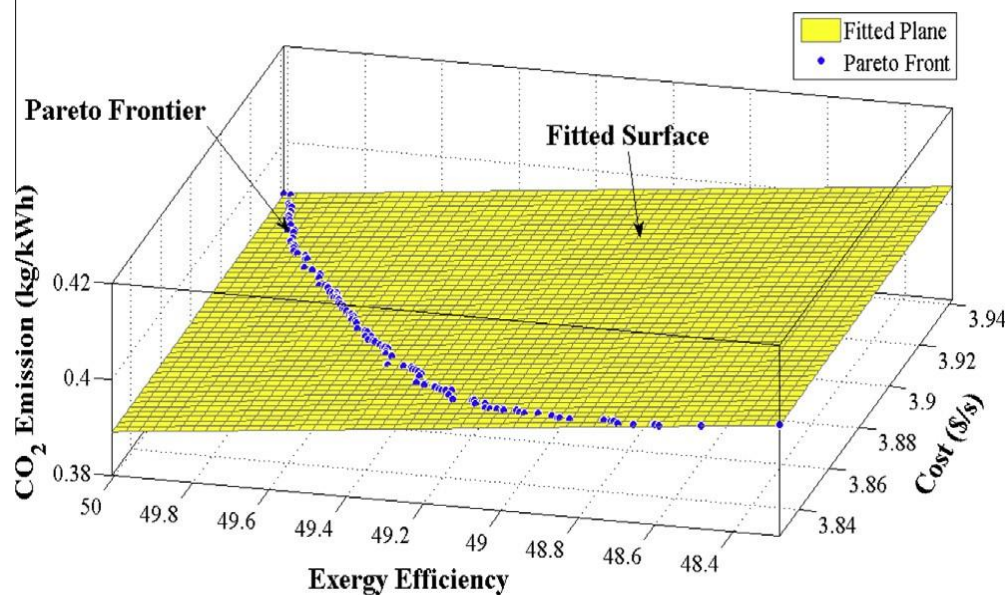


Fig. 3. 3D frontier and plan of feasible solutions for the case of 80% vapor quality at steam turbine outlet.

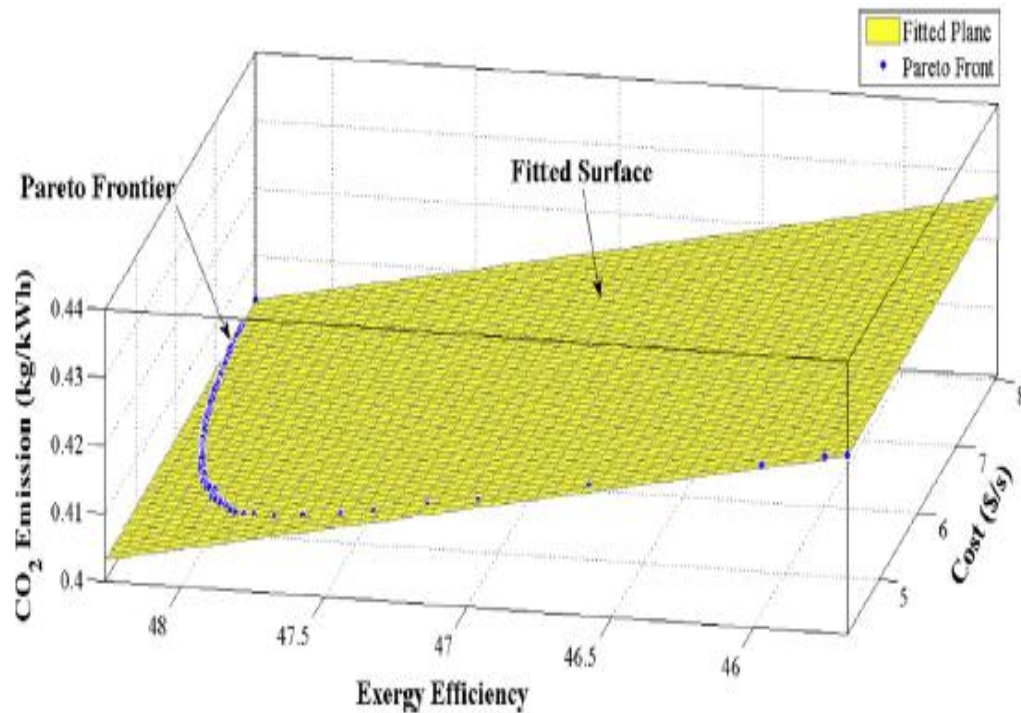


Fig. 4. 3D frontier and plan of feasible solutions for the case of 88% vapor quality at steam turbine outlet.

Table 5

Fitted surface for three mentioned surfaces.

Case	Fitted surface
1	$CO_2 = 0.7601 + 0.01959 \times C_{Total} - 0.814 \times \eta$
2	$CO_2 = 0.7901 + 0.005143 \times C_{Total} - 0.8216 \times \eta$
3	$CO_2 = 0.8298 + 0.0000127 \times C_{Total} - 0.8843 \times \eta$

the three mentioned cases. In this respect, a set of asymptotic are plotted for each Pareto frontier and their intersection shows the equilibrium line. Thus, the best solution is the point which has the minimum distance from the equilibrium point. On the other side, the optimum point seems to be farther than some other points on the curve due to the coordinates are not in the same scales. In addition, the procedure of calculating the equilibrium point and minimum distance point is automatically done in the programme and are applied to all cases.

5.3. Cost constraint

Equilibrium point method leads to the following tabulated performance data for each case as the best case. By considering cost limits and efficiency variations, one may conclude that higher vapor quality at the turbine outlet means higher cost, lower efficiency and also lead to the higher cost variation rate per efficiency improvement. In fact, high vapor quality at the outlet requires employment of better technology in the plant which could not be cost effective. The data of each case as the best case that was calculated using the equilibrium point method is shown in [Table 6](#). By comparing the data calculated from the equilibrium point method with the current design data, it is observed that the cost of third case for its optimal solution is much higher than the original design point. Therefore, this case cannot be a solution due to the cost of employing this case is much higher than the accepted value. On the other hand, if the financial resources are limited to a specific value the method of equilibrium point must be used with a cost constraint. In this case the cost constrains depend on the current cost of the plan. The process of best point selection is shown in [Fig.8](#).

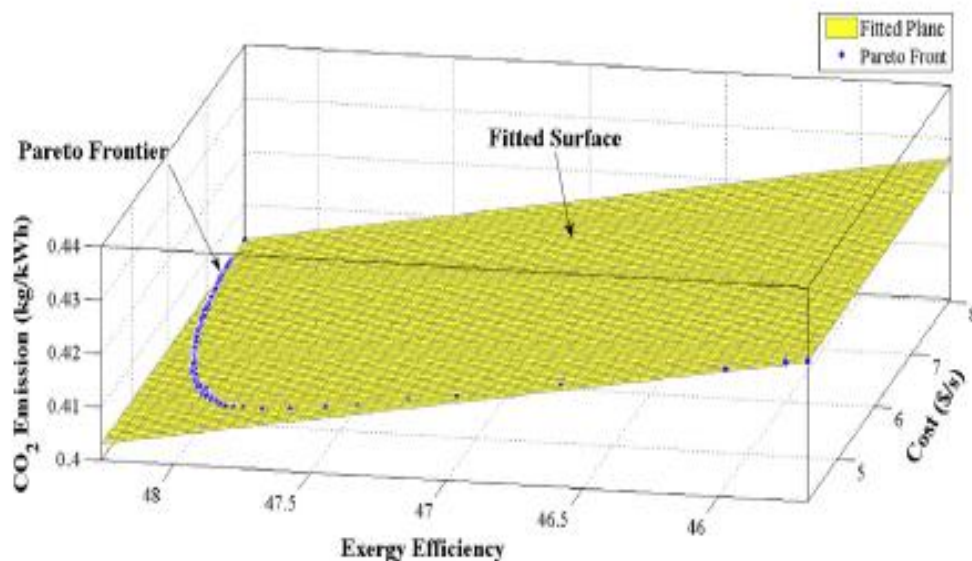


Fig. 4. 3D frontier and plan of feasible solutions for the case of 88% vapor quality at steam turbine outlet.

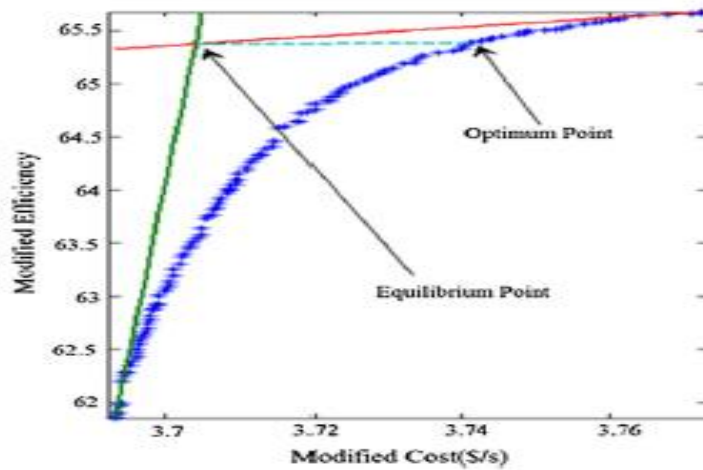


Fig. 5. Equilibrium point and the best solution on the transformed plan of cost efficiency for no quality constraint case.

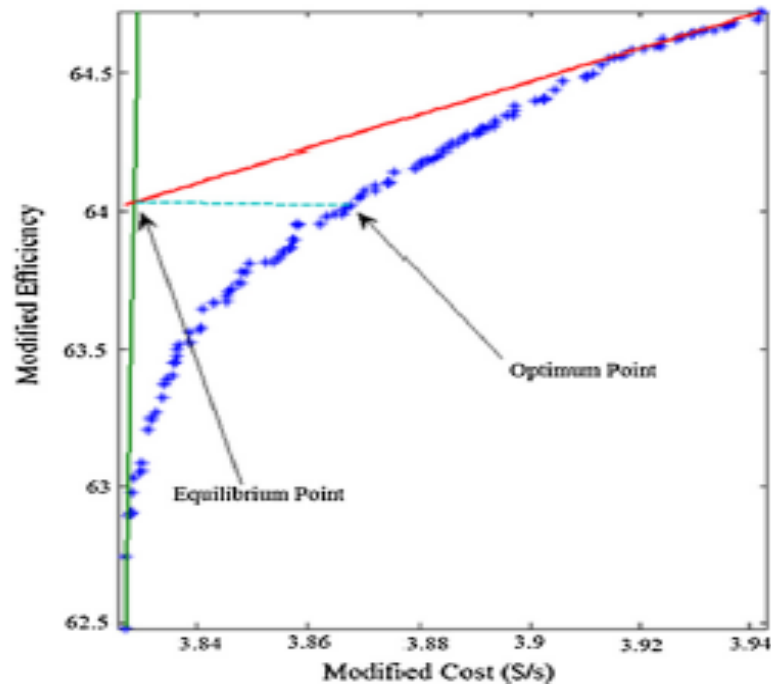


Fig. 6. Equilibrium point and the best solution on the transformed plan of cost efficiency for quality constraint of 80%.

the best solution. Thus, for case 3 the best solution is different and is listed in [Table 7](#). It is clearly shown in [Table 7](#) the cost constraint does not greatly affect the efficiency of the best point. The call off point is the cost that is changed per efficiency improvement is relatively high in case 3 with high value of vapor fraction at the turbine outlet.

5.4 Effect of quality on the cycle performance, economy and environmental footprint

Quality at the steam turbine outlet is a restrictive parameter in the optimization process. To appreciate the effect of this parameter, the data of all the four cases (an extra case regarding to the cost constraint) are presented here. In Fig. 9 performance curve of the steam cycles for mentioned cases are presented. As shown in this figure, an increase in the quality at the turbine outlet leads to increase the temperature and pressure at steam turbine inlet. In the optimization study, the optimum data tends to reach the lower qualities due to the fact that the condenser rejected heat tends to be as slow as possible. Consequently, an efficient two phase flow.

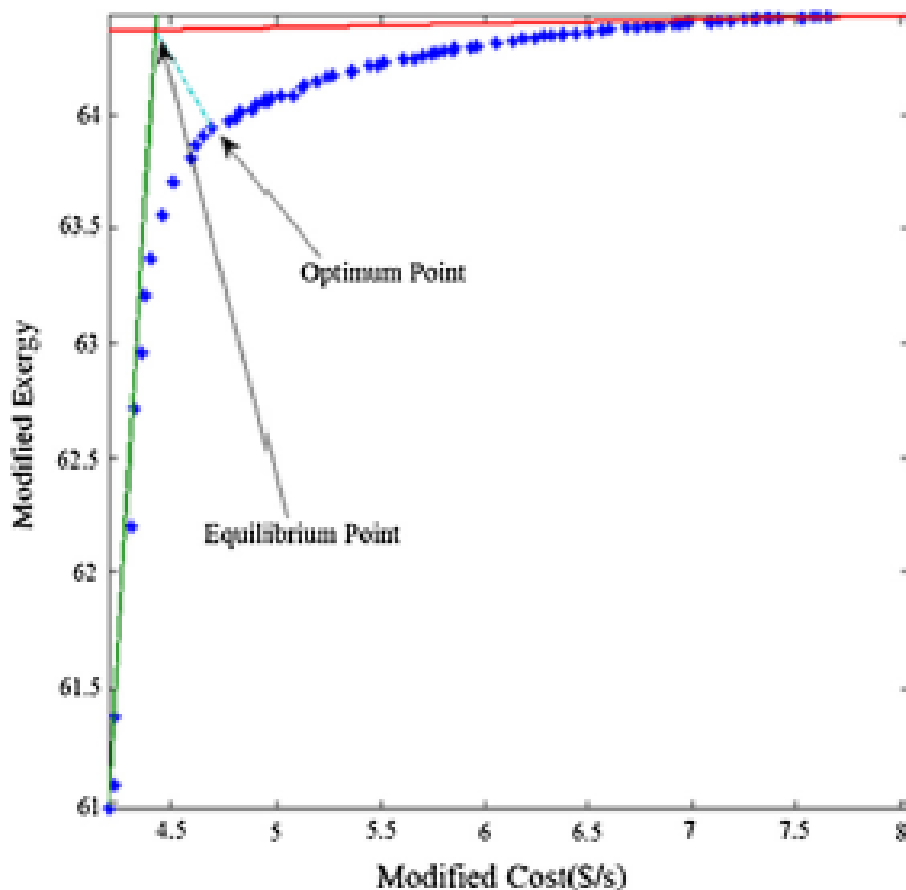
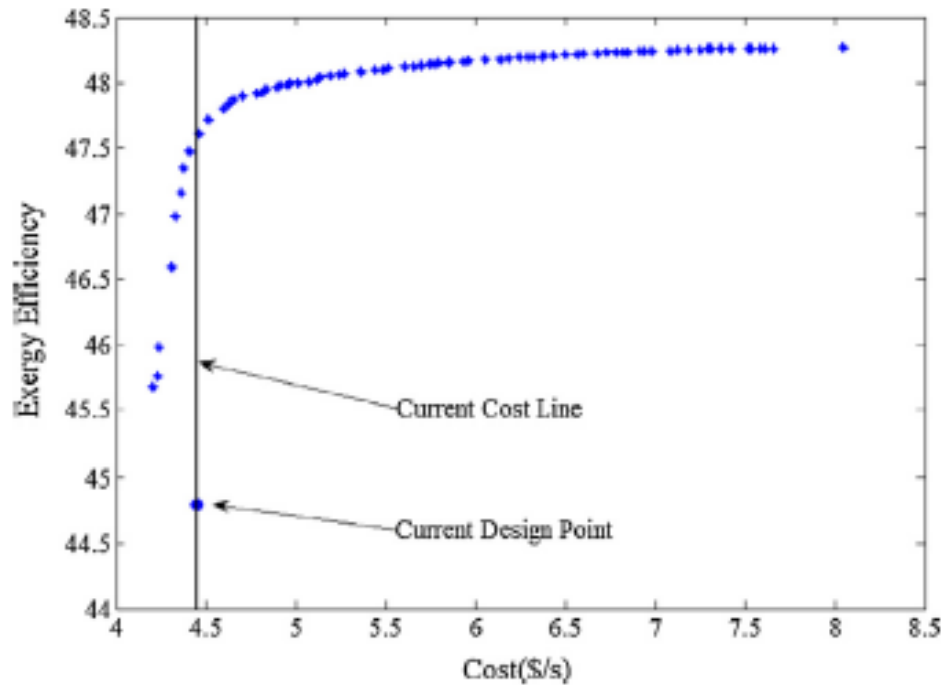


Fig. 7. Equilibrium point and the best solution on the transformed plan of cost efficiency for quality constraint of 88%.

Table 6

Best Solutions for the equilibrium point selection method.

Case	Cost (\$/s)	Efficiency (-)	Emission (kg/kW h)
1 (no constraint)	3.7	0.486	0.401
2 (quality = 0.8)	3.841	0.492	0.396
3 (quality = 0.88)	6.325	0.482	0.404
Design point	4.45	0.448	0.435

**Fig. 8.** Cost constrain and the best solution.**Table 7**

Best point using equilibrium point and cost constraint approach for case 3.

	Cost (\$/s)	Efficiency	CO ₂ (kg/kW h)
3 (quality = 0.88) (equilibrium method)	6.325	0.482	0.404
3 (quality = 0.88) (cost constraint)	4.412	0.475	0.410
Design point	4.45	0.448	0.435

heat transfer phenomena is dominated in the HRSG when lower degrees of superheat is required. Before applying the cost constraint, pressure in the LP line decreases due to the lower pressure and low temperature are needed in HRSG. Consequence, lower superheat degree at the stage inlet and outlet of the steam turbine is occurred which leads to decreases condenser rejected heat as disused earlier. But by applying cost constraint, the highest temperature and pressure of the cycle increase as illustrated in Fig. 9. In Fig. 10 the effects of quality control on the optimum gas cycle are presented. To produce a higher temperature steam, the turbine inlet and outlet temperatures need to be high enough. An increase in gas turbine inlet temperature and also combustion chamber working temperature, results in an increase in the cost

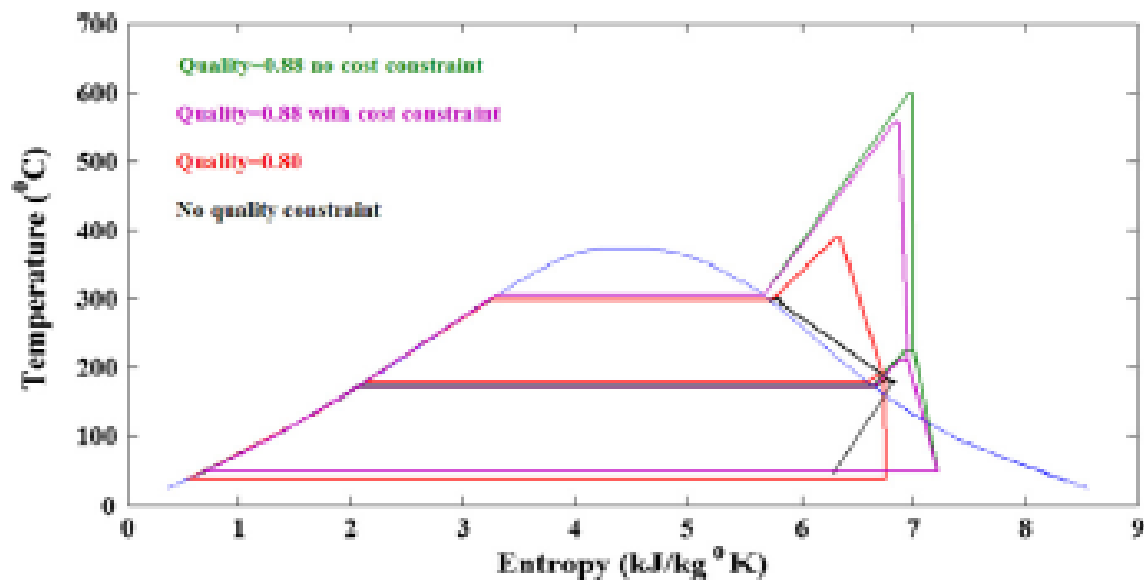


Fig. 9. T-S diagram of steam cycle for different cases.

of components. When cost constraint is applied, the cycle performance changes significantly. Lower isentropic efficiency of the compressor leads to a lower compressor cost. Moreover, by applying cost constraint, a fuel flow reduction occurred in the duct burner which leads to reduce the fuel cost, HRSG cost, duct burner cost and also the cost of steam cycle. Thus, cost constraint reduces the overall cost significantly while the steam cycle reaches higher quality.

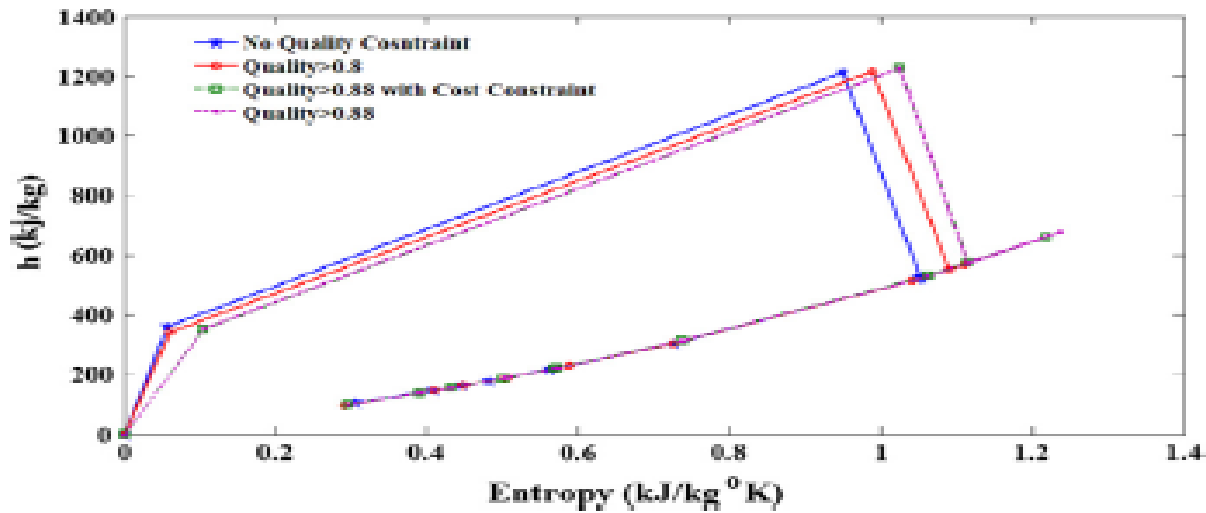


Fig. 10. h - S diagram of gas cycles for different cases.

Furthermore, the components efficiency of the gas cycle will be lower and entropy changes in the gas cycle are higher. Fig. 11 provides a comparative presentation for the decision variables. In this graph, each decision variable is normalized by its maximum value for 4 cases. As it is obvious in this figure, for the steam quality outlet of 88%, the compressor efficiency and compressor pressure ratio are significantly lower which causes lower cost and higher gas cycle outlet temperature. Steam turbine efficiency has the highest value for the quality of 88% outlet with no cost constraint due to high superheat degree at the inlet. The high superheat degree in steam turbine inlet allows higher turbine efficiency to be selected regardless of the quality constraint at outlet. Duct burner fuel mass flow rate is the lowest for the case of no constraint due to the fact that, chemical reactions are the most destructive phenomenon. An increase in fuel mass flow rate causes extra cost flow and extra emission in addition to the gas cycle. So, if there is no constraint, less fuel will be burnt in the duct burner which decreases the cycle efficiency. However, the exergy of exhaust gases extracted only to produce more work. Therefore, this system with 88% quality at steam turbine outlet and cost constraint is the most efficient, economical and environmental friendly.

Mass flow rate of HP steam increases when the steam quality reaches the higher

values but LP steam mass flow rate is reduced. In the same manner, pinch point temperature of LP drum increases while HP drum pinch point reduces. In general when quality is under control and needs to be high, the optimization tool tries to find cycles with higher bottoming cycle efficiency.

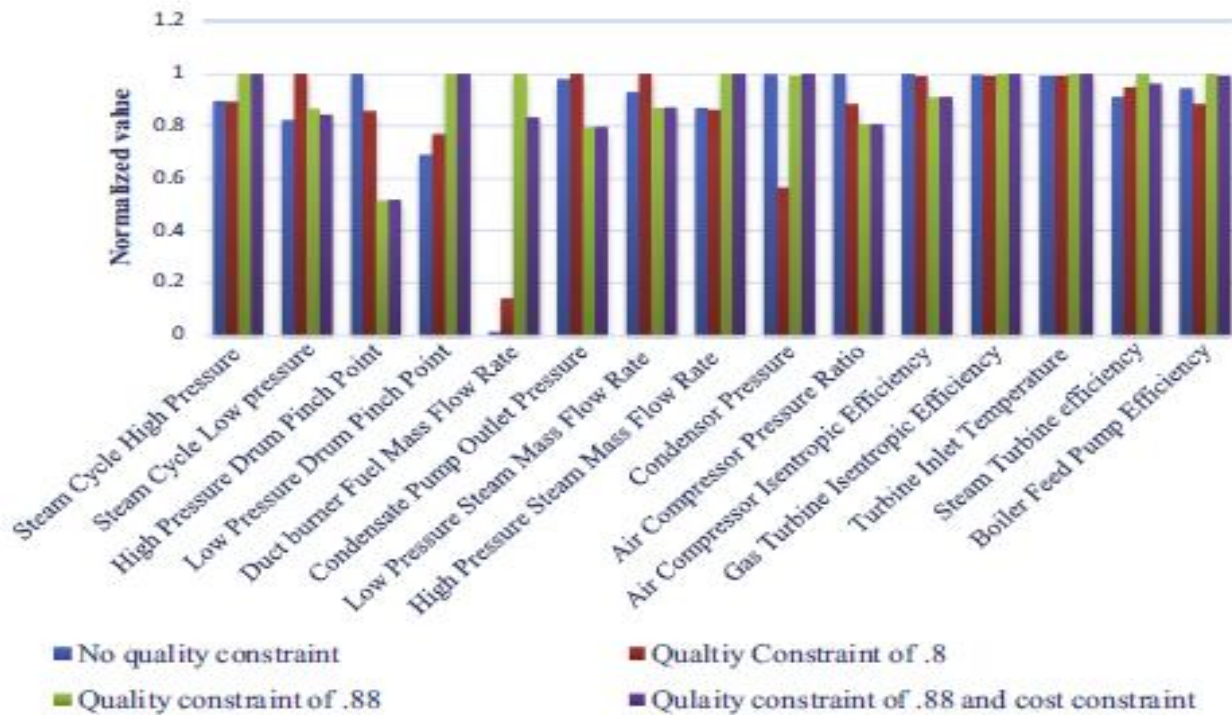


Fig. 11. Comparative presentation of decision variables for different case.

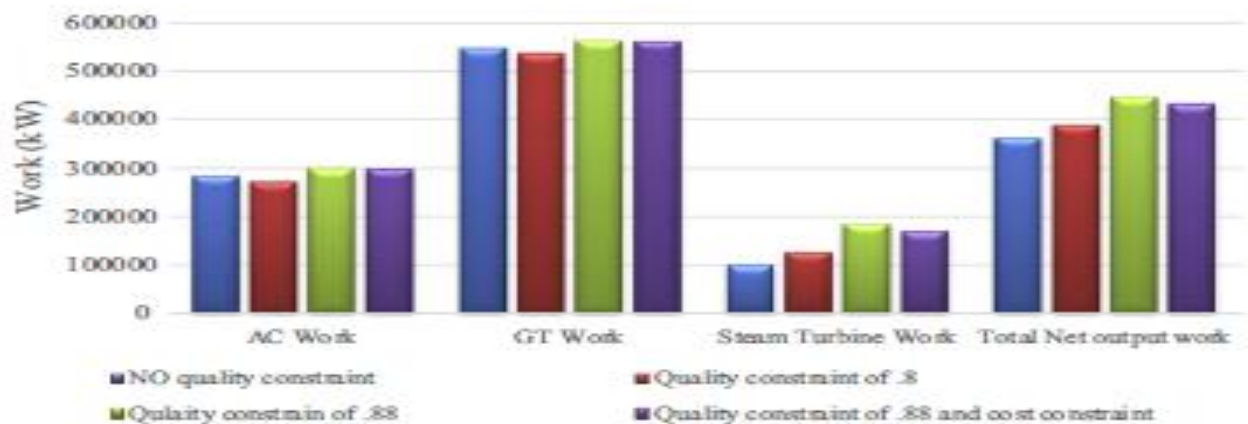


Fig. 13. Works flows in the cycles for different cases.

steam cycles falls sharply which means lower steam cycle and overall efficiency. Fig. 13 shows the variation of works flows in different cases. In higher quality values, the system in optimum point leads to higher works flows and thus, net power output increases significantly. There as on of higher works flows is that the higher degree of superheat, more hot gas flow, higher HRSG gas inlet temperature and higher steam turbine efficiency will applied. Consequence, the net power output of the system especially in the steam cycle is increased.

Exergy efficiency of different components with different qualities are compared and presented in Fig. 14. It is obvious that an increase in the steam turbine outlet quality leads to a sharp decrease in exergy efficiency of the compressor. Combustion chamber and gas turbine efficiencies are kept approximately constant. Duct burner efficiency decreases due to the higher fuel mass flow rate and temperature gradient in the duct burner. HRSG components reached higher efficiencies and also steam turbine components reached higher values. Generally, cycle efficiency decreases due to the efficiency of HRSG and steam cycle efficiency does not increase significantly. By applying cost constraint, the efficiency of steam cycle and bottoming cycle decrease. Moreover, total efficiency of optimum cycles fall slightly as previously compared.

The purchase costs of the power plant components are presented in. They are normalized with the maximum value in three cases with different qualities and also one case with cost constraint, this figure provides a comparative method of system

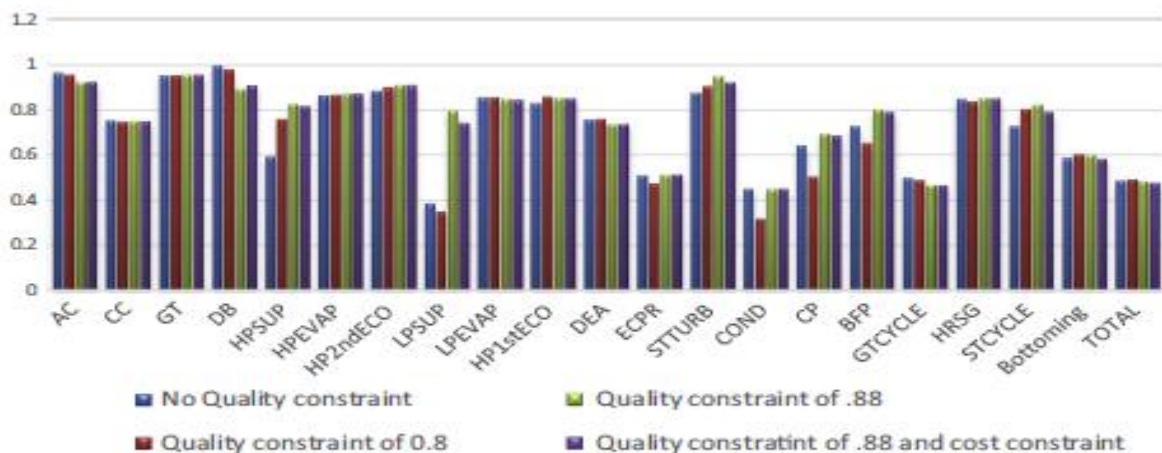


Fig. 14. Exergy efficiency of different components and subsystems.

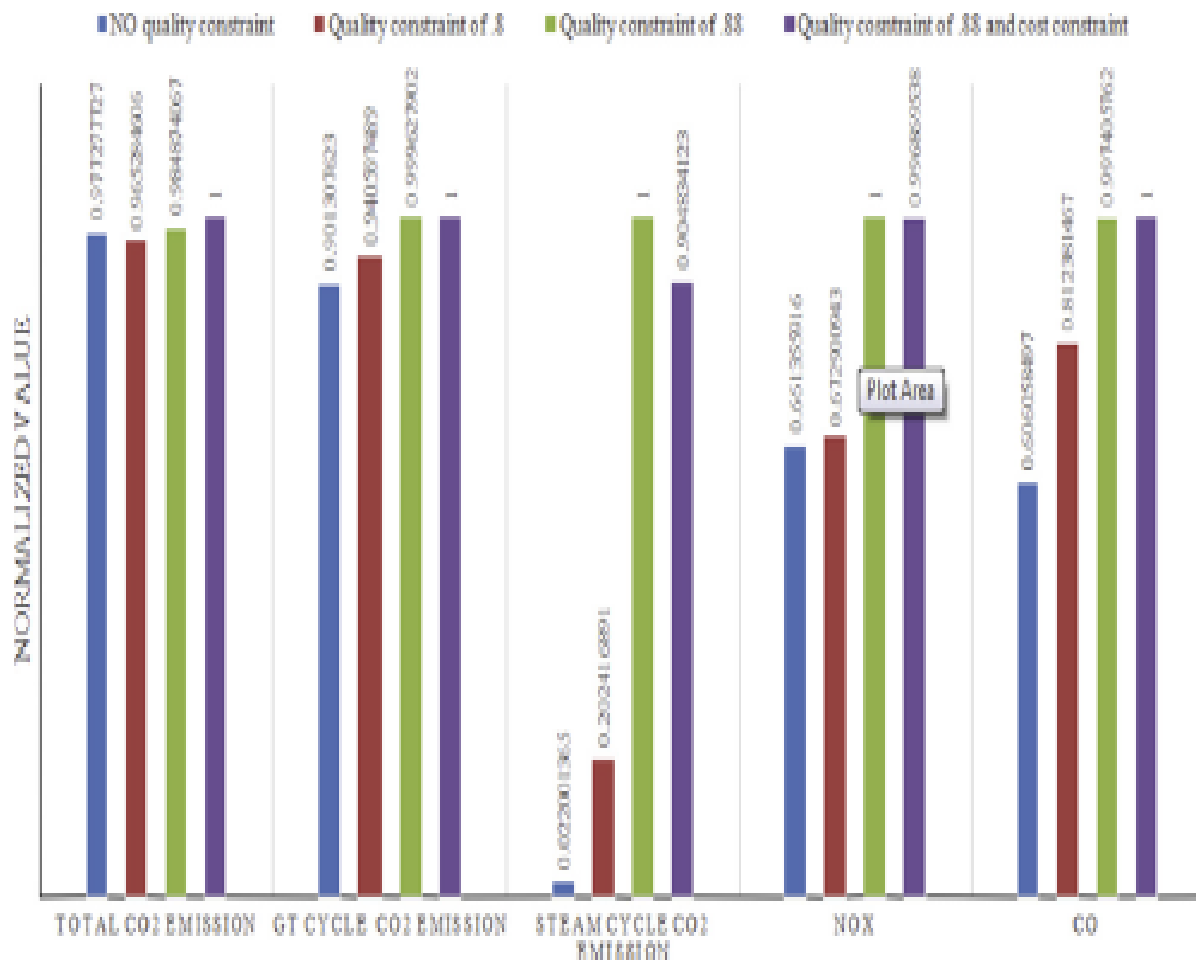


Fig. 16. Comparative analysis of emission for different quality constraints values.

Performance investigation. Highest values costs are generally for the cases with higher quality at outlet. There are some exemptions like air compressor which are less prcey when quality is lower. Comparing the cost constraint case with quality outlet only case 88% qualities outlet tells which components are affected to reach lower purchase costs. Variation in gas turbine cycle purchase cost are significantly lower than steam cycle and bottoming purchase cycle costs.

CHAPTER -6

6. Analyzing and comparing basic design and optimum cycle data

In this section, the basic design data is compared with the optimum cycle data which was concluded from the optimization results. Therefore, the data of decision variables for basic design and optimum point are listed in Table 8. In order to compare the cycles in an easy manner, variation of the design variables from actual data is shown in Fig. 17. As it is clear in this figure, gas turbine cycle parameters increase except compressor efficiency. Steam cycle mass flow rate decreases especially for low pressure mass flow rate while the efficiencies of the components in bottoming cycle are increased.

Table 8
Decision variables for design and optimum cycles [1].

Decision variable	Optimized	Design	Unit
Steam Cycle High Pressure	103.542	110.460	Bar
Steam Cycle Low Pressure	9.012	11.280	Bar
High Pressure Drum Pinch Point	2.537	4.000	K
Low Pressure Drum Pinch Point	22.800	24.500	K
Duct burner Fuel Mass Flow Rate	0.804	0.800	Kg/s
Condensate Pump Outlet Pressure	2.497	5.000	Bar
Low Pressure Steam Mass Flow Rate	12.904	18.000	Kg/s
High Pressure Steam Mass Flow Rate	132.347	134.200	Kg/s
Condenser Pressure	0.112	0.090	Bar
Air Compressor Pressure Ratio	11.066	10.100	(-)
Air Compressor Isentropic Efficiency	0.808	0.864	(-)
Gas Turbine Isentropic Efficiency	0.903	0.881	(-)
Turbine Inlet Temperature	1336.780	1244.150	K
Steam Turbine Efficiency	0.898	0.840	(-)
Boiler Feed Pump Efficiency	0.872	0.750	(-)

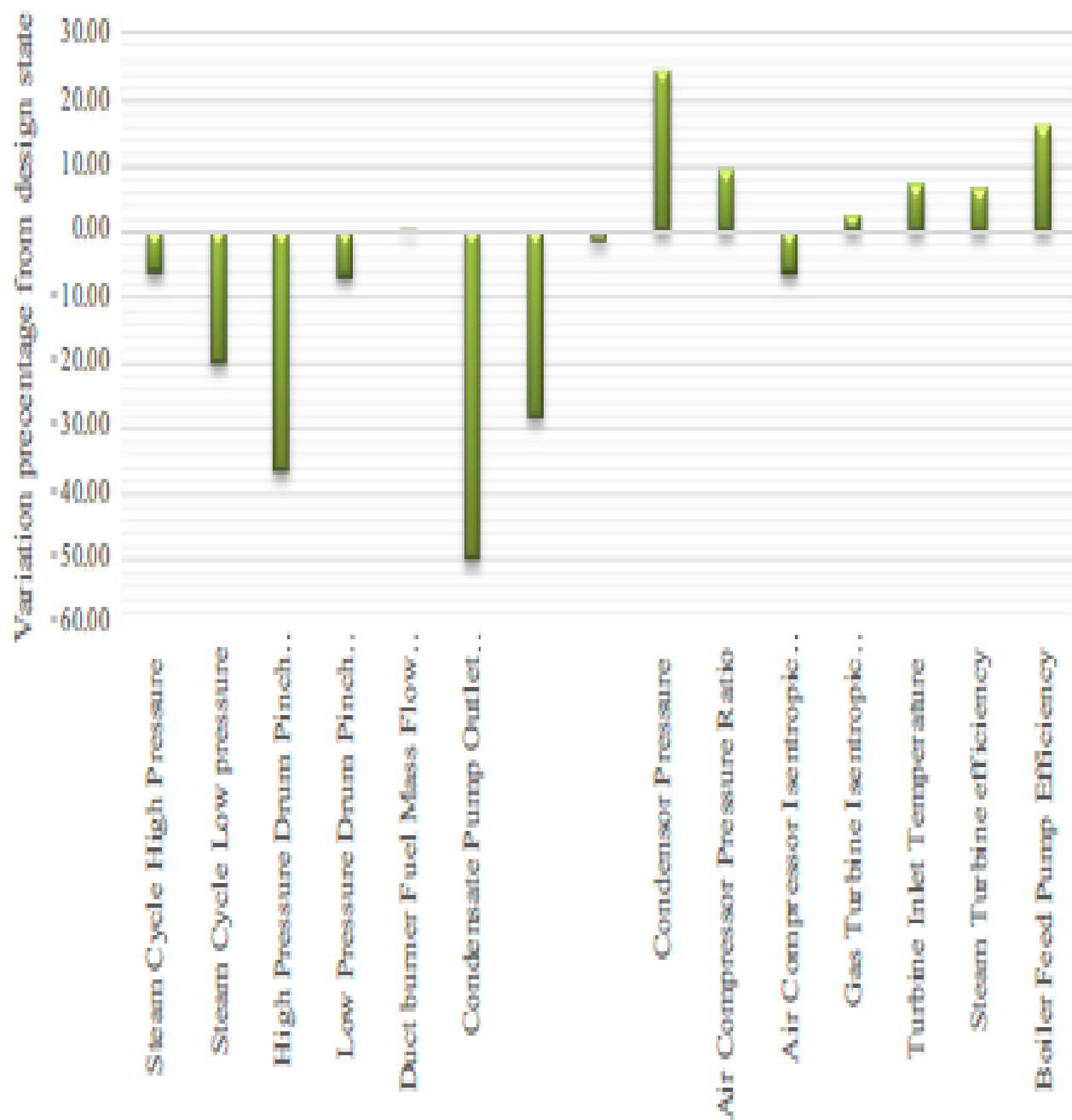


Fig 17. Changes in variables after optimization.

7. METHODS TO IMPROVE EFFICIENCY OF CCGT

7.1 Boiler Efficiency:

Efficiency can be determined as Ratio of output to input.

We are giving Heat as Boiler input & steam as boiler output.

The efficiency can be calculated by using two methods

- 1) Direct Method
- 2) Indirect Method.

7.2 Direct Method

This is also known as “**input-output method**” due to the fact that it requires only the useful steam output and the heat input by fuel or flue gas for finding the efficiency.

$$\eta = \frac{Q_{out}}{Q_{in}}$$

$$\eta = \frac{q (H - h_f)}{m_g c_p (t_{f_g in} - t_{f_g out})}$$

Q_{out} = Output from boiler

Q_{in} = Input to boiler

q = Heat in steam kj/s

H = Enthalpy of steam

h_f = Enthalpy of feed water

$t_{f_g in}$ = Temperature of hot flue gas entering the boiler

$t_{f_g out}$ = Temperature of flue gas leaving the boiler

m_g = Mass of flue gas in kg/sec

c_p = Specific heat of flue gas in kj/kgk

Based on design Parameters

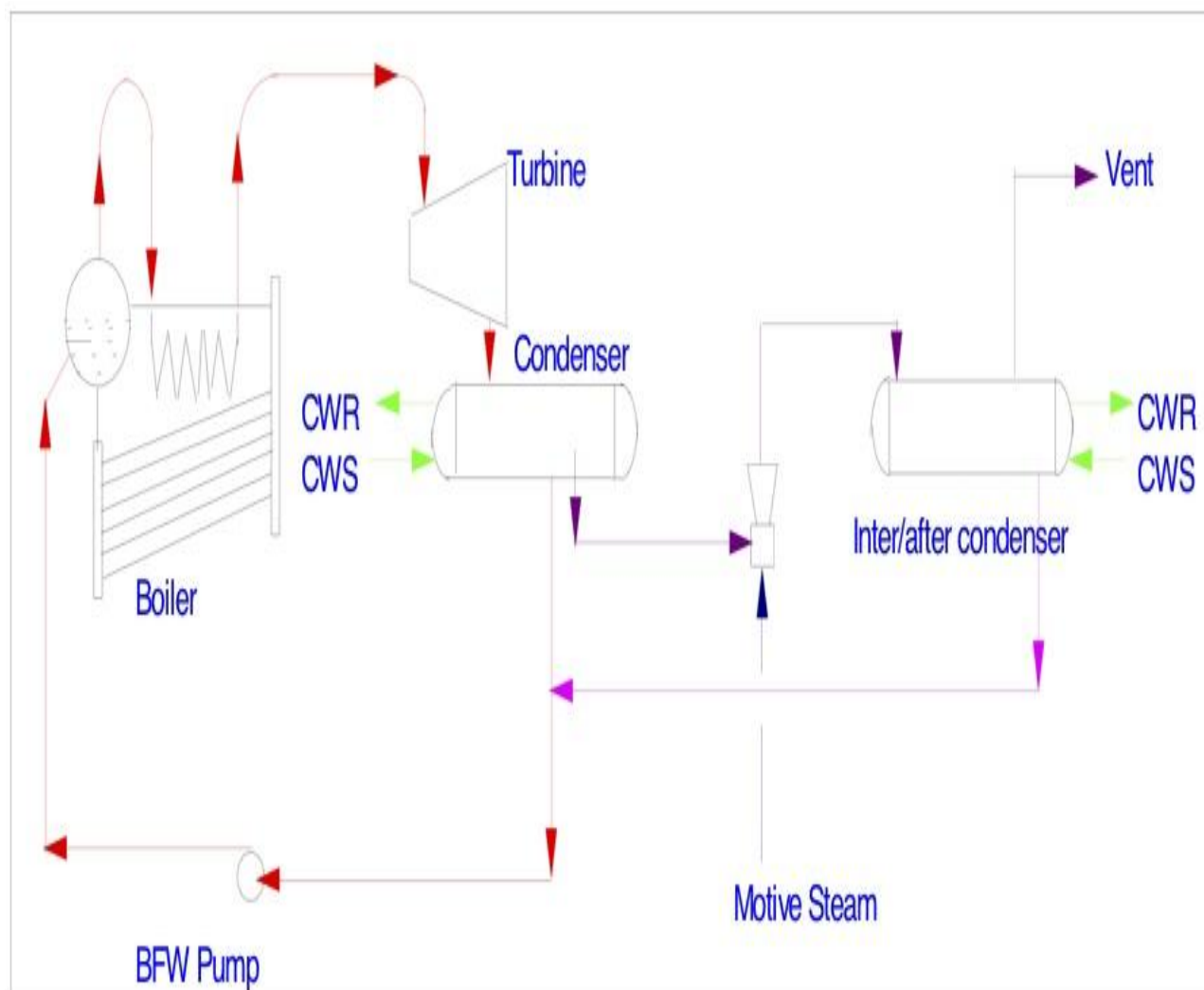
	WHRB#1	WHRB#2	WHRB#3	WHRB#4
Steam Temp.	325	325	325	325
Steam Pressure	18	18	18	18
Steam Flow	10	12.94	9.1	20.6
Feed water Temp.	130	130	130	130
Feed water press.	25 to 26 kg/cm ²			
Enthalpy of Steam (kcal/kg)	738	738	738	738
Enthalpy of Feed water (kcal/kg)	131	131	131	131
Flue gas flow(N M ³ /hr)	100000	137000	111000	250000
flue gas density	1.4	1.4	1.4	1.4
Sp. Heat of flue gas	0.24	0.24	0.24	0.24
Flue gas inlet temp. °C	400	385	365	365
Flue gas outlet temp. °C	240	241	240	240
Total heat input in boiler(kcal/hr)	5376000	6628608	4662000	10500000
heat out from boiler	6070000	7854580	5523700	12504200
boiler efficiency	113	118	118	119

	WHRB#5	
Steam Temp.	331	
Steam Pressure	15.09	
Steam Flow	12.21	
Feed water Temp. at Eco. inlet	130	
Feed water Temp. at Eco. Out let	160	
Feed water press.	25 to 26 kg/cm ²	
Enthalpy of Steam (kcal/kg)	743	
Enthalpy of Feed water at Eco. inlet (kcal/kg)	131	
Enthalpy of Feed water at Eco. Outlet (kcal/kg)	161	
	Duct- 1	Duct- 2
Flue gas flow(M ³ /hr)	371663	40858
flue gas density	0.48	0.42
Sp. Heat of flue gas	0.24	0.24
Flue gas temp. °C I/L	349	349
Flue gas temp. °C O/L	93	93
Total heat input in boiler(kcal/hr)	10960788	1054332.5
heat out from boiler	11017530	
boiler efficiency	91.70	

7.3 Steam Turbine Efficiency:

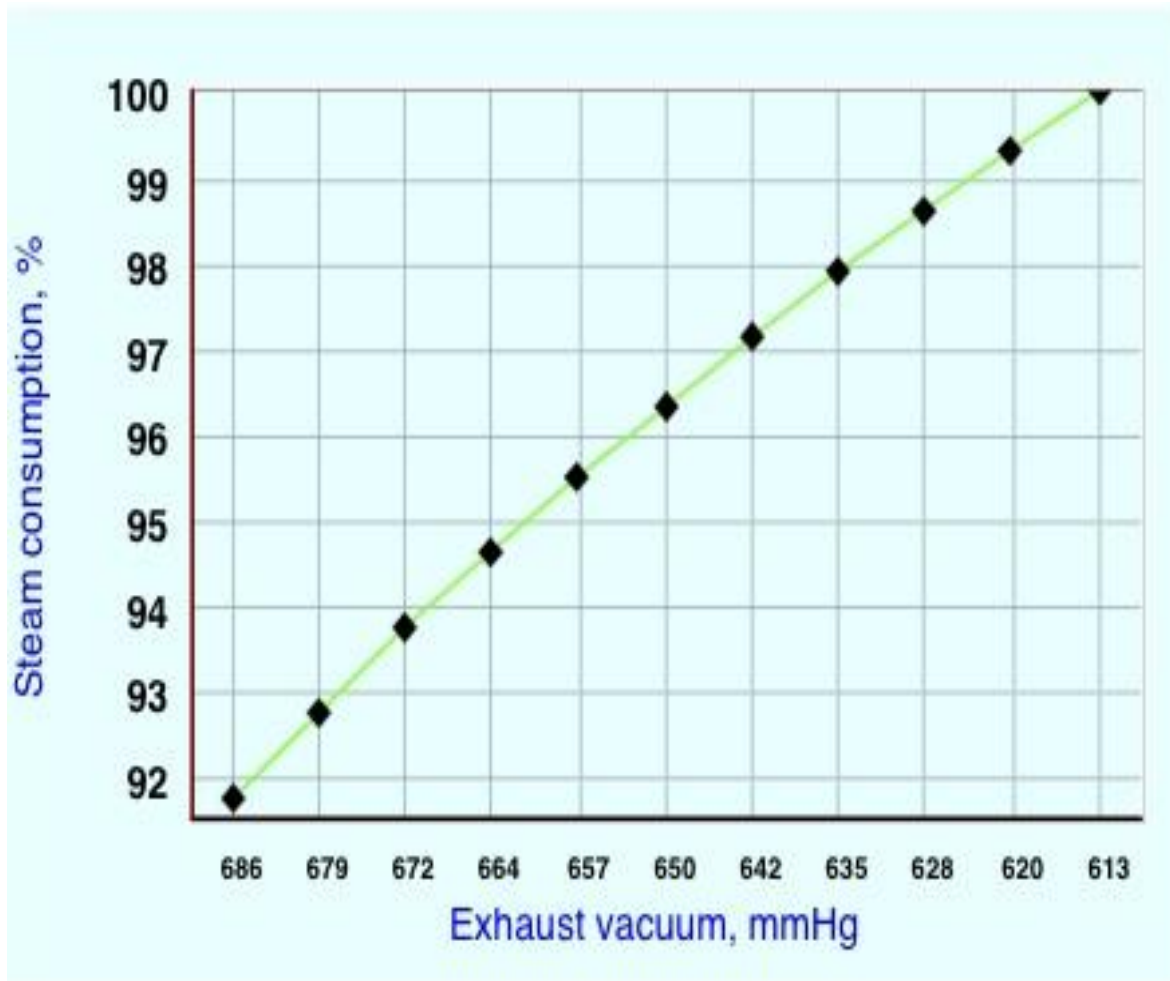
Effect of operating conditions on steam turbines

A condensing turbine system is shown in figure . Turbine exhaust operating below atmosphere, is condensed in a shell and tube exchanger called surface condenser. Condensate flows in the shell side of the condenser and steam is condensed by the cooling water. Vacuum in the surface condenser i.e. turbine exhaust vacuum is controlled/ maintained by vacuum ejector system of the Surface condenser.

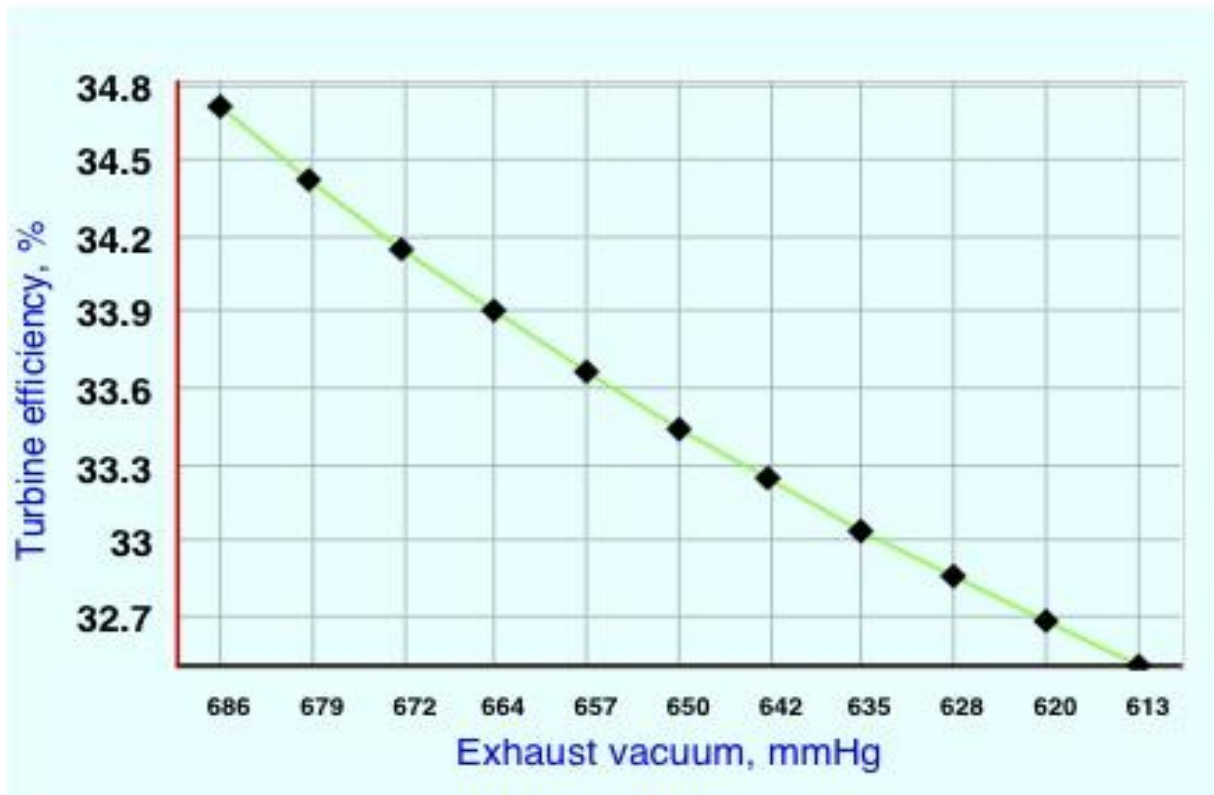


7.4 Effect of exhaust pressure/ vacuum

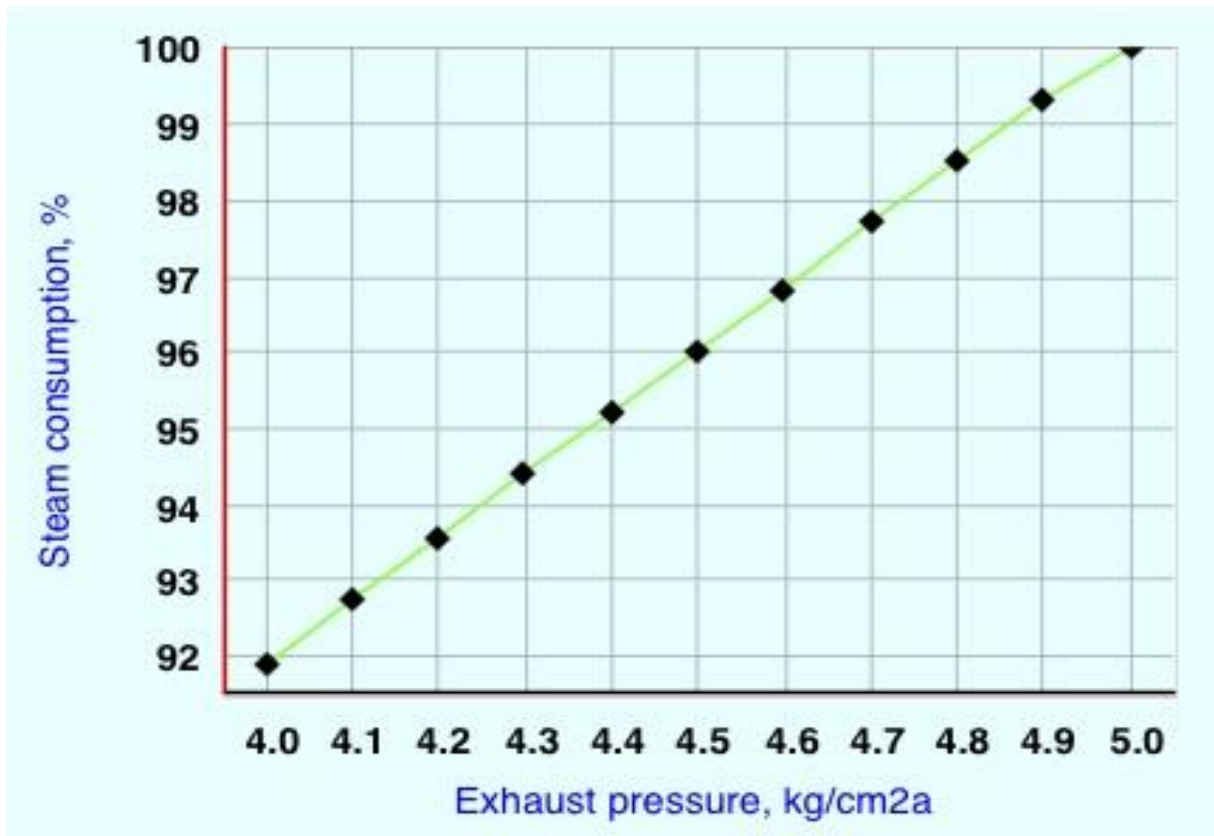
Higher exhaust pressure/ lower vacuum, increases the steam consumption in the turbine, keeping all other operating parameters constant. Exhaust pressure lower than the specified will reduce the steam consumption and improves the turbine efficiency. Similarly exhaust vacuum lower than the specified, will lower the turbine efficiency and reduces the steam consumption. Figure 6a & 6b represents the effects of exhaust vacuum on steam consumption and turbine efficiency respectively, keeping all other factors constant for the condensing type turbine.



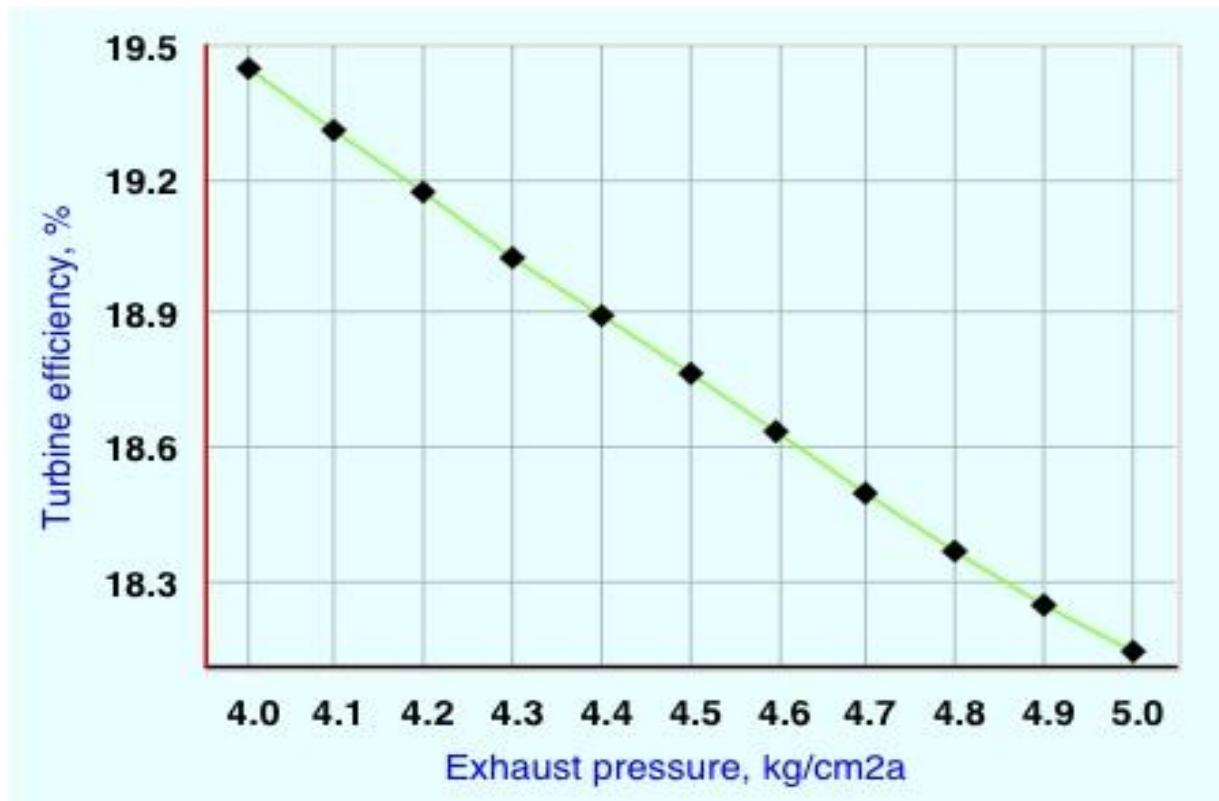
Effect of exhaust vacuum on steam consumption in condensing type turbine



Effect of exhaust vacuum on turbine efficiency in condensing type turbine



Effect of exhaust pressure on steam consumption in back pressure type turbine



Effect of exhaust pressure on turbine efficiency in back pressure type turbine

7.5 Overall thermal efficiency of combined cycle power plants - a sample CCGT presented

Calculating or predicting the overall performance of a combined cycle power plant, specifically a combined cycle gas turbine (CCGT) power plant is sometimes difficult for most design engineers. Your favorite energy technology expert again comes to the rescue - Engineer Marcial T. Ocampo - has derived the following equation to guide the design engineer and project finance modeler or business development engineer in predicting the overall thermal efficiency of the combined cycle.

Here's the step by step derivation:

Let Total Energy Input = Total Energy Output = 100%.

Then let's assume that the proportion of the energy input going to the gas turbine (GT) and to the waste heat recovery steam generator (SG) is around:

$$GT = 1/3 = 33.33\%$$

$$SG = 1 - 1/3 = 2/3 = 66.67\%$$

The energy input to the steam generator is then split into 85% (up to 90%) being recovered as steam energy in the boiler while the balance of 15% (or 10%) is lost to the atmosphere (smoke stack, radiation and convection loss).

The steam energy is then captured in the steam turbine with an efficiency of 37% and the balance (63%) is lost to the cooling water in the condenser.

The mechanical energy captured in the GT and ST shafts are then coupled in a clutch (100% efficiency) to drive finally the electric generator (EG) with mechanical to electrical conversion efficiency of 98% (balance of 2% lost to heating in the generator windings and generator cooling system).

The final formula is thus: **OE = (GT x GTE + SG x BE x STE) x ME x GE**

Where OE = overall energy efficiency, % of fuel energy input (as GHV or LHV)

$$GT = 1/3 = 33.33\% \text{ (assumption)}$$

$$GTE = \text{gas turbine efficiency} = \text{say } 90\% - 95\%$$

= function of heat loss from GT due to conduction, convection and radiation from GT casing, any heat loss due to friction resulting in higher GT exhaust is captured, however, by the heat recovery steam generator

$$SG = 1 - GT = 1 - 1/3 = 2/3 = 66.67\% \text{ (balance)}$$

$$BE = \text{waste heat recovery boiler efficiency} = 85\% \text{ (up to } 90\%)$$

= function of exit flue gas temperature (energy lost), gas turbine exhaust temperature (energy input), boiler design, flue gas composition (fuel, excess air).

$$STE = \text{steam turbine efficiency} = 37\% \text{ (up to } 40\%)$$

= function of steam inlet temperature, steam exhaust pressure, condenser vacuum pressure,

cooling system, steam turbine design, steam quality

ME = mechanical drive shaft and clutch efficiency = 100%

= function of drive shaft design, clutch system, bearing lubrication, mechanical design,

Windage losses (air drag)

GE = electric generator efficiency = 98%

= function of generator design, voltage, windage losses (air drag), type of gas cooling

(hydrogen, air), bearing lubrication.

Putting all together now, the predicted overall thermal efficiency of the combined cycle power plant is:

$$OE = (33.33 \times 95\% + 66.67\% \times 85\% \times 37\%) \times 100\% \times 98\% = 51.58\%$$

For instance, if we raise the gas turbine efficiency from 95% to 98%, the resulting overall efficiency would be raised to

$$OE = (33.33 \times 98\% + 66.67\% \times 85\% \times 37\%) \times 100\% \times 98\% = 52.56\%$$

For instance, if we raise the boiler efficiency from 85% to 90%, the resulting overall efficiency of the combined cycle power plant is raised to:

$$OE = (33.33 \times 98\% + 66.67\% \times 90\% \times 37\%) \times 100\% \times 98\% = 53.77\%$$

Further raising the steam turbine efficiency to 40% will result in a much higher efficiency of:

$$OE = (33.33 \times 98\% + 66.67\% \times 90\% \times 40\%) \times 100\% \times 98\% = 55.53\%$$

I guess this provides an upper limit of what a CCGT could deliver, around 56%.

The only way to go higher than this is to improve further the gas turbine efficiency beyond 98%, boiler efficiency beyond 90%, raising the steam turbine efficiency beyond 40% and optimizing the proportion of energy output thru the gas turbine (currently 1/3) and the steam generator (balance of 2/3).

The author has developed an state of the art project finance model for a 25-year economic life CCGT and may be requested thru this link:

The model has the following capability:

1) Given the all-in capital cost (EPC, installation and erection, taxes and duties, project development, regulatory costs, working capital, interest during construction), O&M costs (variable and fixed O&M, recurring regulatory costs, property taxes, ROW and land lease, property insurance, business interruption insurance, etc) and electricity tariff (industries, distribution utilities, national grid, wholesale electricity spot market) ==> it determines the maximum price of the natural gas fuel needed to meet the 15% p.a. DCF IRR for equity investment.

2) Alternatively, if the capital cost, O&M costs, electricity tariff and fuel cost (natural gas, gas oil, bunker oil) are fixed, the model will calculate the net present value and project IRR (return of investment or ROI from the project cash flow), NPV and equity IRR (return on equity or ROE from the equity cash flow) or the more stringent financial analysis tool of bankers today --> the NPV and dividends IRR (return on dividends from dividends cash flow).

Other variations of the model objective include determining the maximum capital cost given the O&M cost, tariff and fuel cost to meet the 15% equity IRR hurdle rate for making equity investments. The model is currently capable of analyzing several CCGT machines / models from various manufacturers for a number of plant locations with specific type of cooling system (sea water once thru, lake water once thru, river water or deep well cooling tower, and dry cooling with radiator).

By using case switches, the model will go over each pair of sensitivity (machine and plant location/type of cooling) and calculate the fuel cost needed to meet 15% equity IRR. It uses goal seek to set the equity cash flow NPV to zero for each of sensitivity pair.

7.6 CONDENSER:

A **surface condenser** is a commonly used term for a water-cooled shell and tube heat exchanger installed on the exhaust steam from a steam turbine in thermal power stations. These condensers are heat exchangers which convert steam from its gaseous to its liquid state at a pressure below atmospheric pressure. Where cooling water is in short supply, an air-cooled condenser is often used. An air-cooled condenser is however, significantly more expensive and cannot achieve as low a steam turbine exhaust pressure (and temperature) as a water-cooled surface condenser. Surface condensers are also used in applications and industries other than the condensing of steam turbine exhaust in power plants.

PURPOSE:

In thermal power plants, the purpose of a surface condenser is to condense the exhaust steam from a steam turbine to obtain maximum efficiency, and also to convert the turbine exhaust steam into pure water (referred to as steam condensate) so that it may be reused in the steam generator or boiler as boiler feed water.

The adjacent diagram depicts a typical water-cooled surface condenser as used in power stations to condense the exhaust steam from a steam turbine driving an electrical generator as well in other applications. There are many fabrication design variations depending on the manufacturer, the size of the steam turbine, and other site-specific conditions.

Steam power plants are still the backbone of the total power generation in the Asia pacific. Thus even a small improvement in the form of increasing the efficiency has a tremendous effect on the fuel saving and also reduction in emission of green house gases. Thus one should not miss out any opportunity to find out the ways and means to increase the efficiency of the steam power cycle. The ideal behind any improvement or modification is to increase the thermal efficiency of the power plant. Thus thermal efficiency improvement techniques are:

- By decreasing average temperature at which heat is rejected from the working fluid (steam) in the condenser. (Lowering condenser Pressure).

Diagram of water-cooled surface condenser

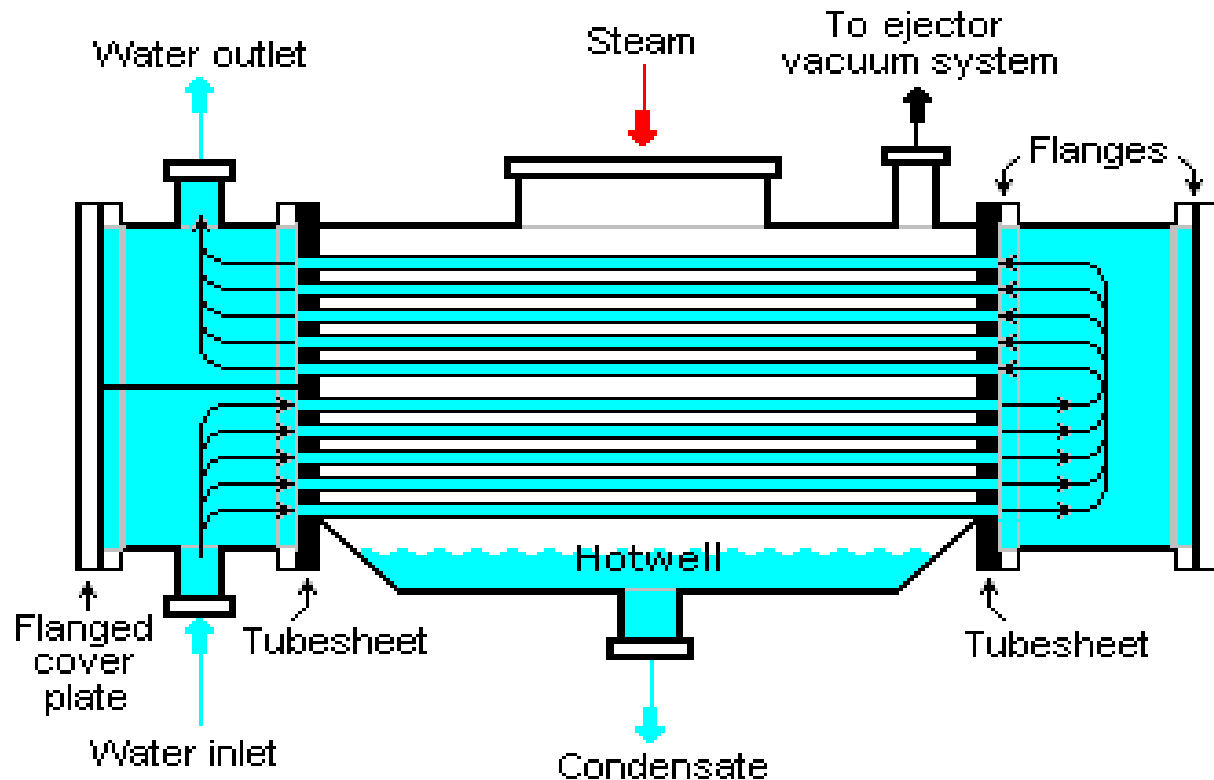


Diagram of a typical water-cooled surface condenser

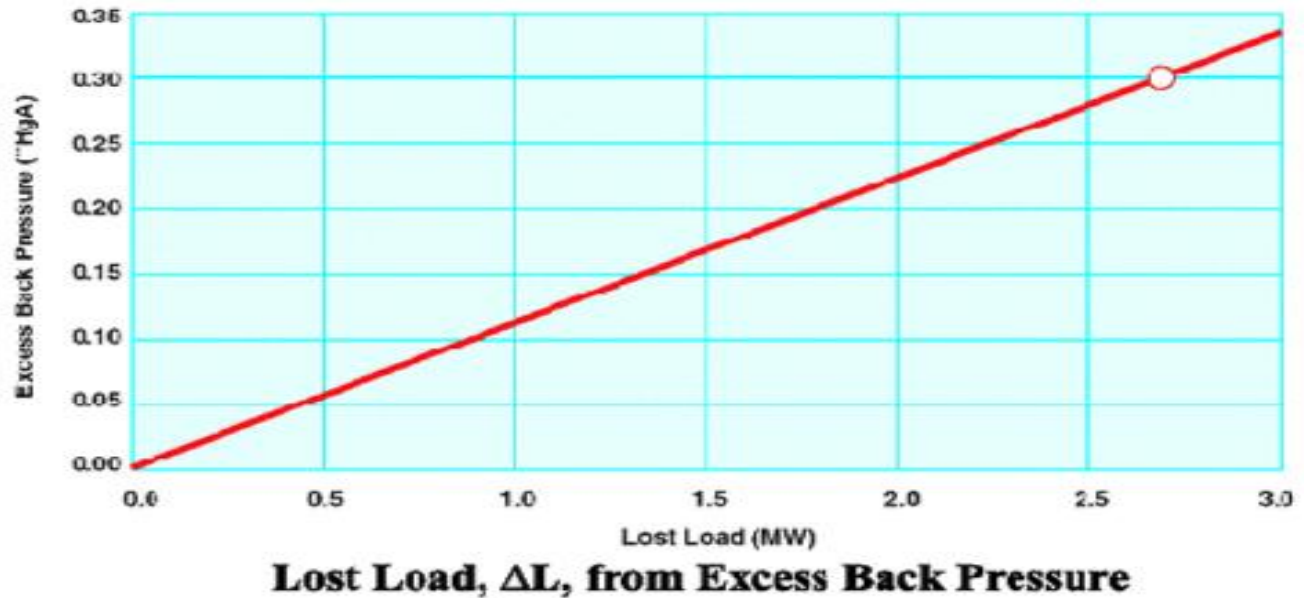
Lowering the Condenser Pressure

Steam leaves the turbine and enters the condenser as saturated mixture in line with the corresponding pressure of steam in the condenser. Lowering the condenser pressure always helps in delivering more net work in the turbine as more expansion of steam in turbine is possible. By the help of T-s diagram the effect of lowering the condenser pressure on the performance of the cycle can be seen and understood.

Positive Effects of Lowering the Condenser Pressure

To milk the advantage of higher efficiency Rankine Cycle has to operate on lower condenser pressure usually below atmospheric. But the limit for lower condenser-pressure is defined by the cooling water temperature corresponding to saturation-pressure of the area. In the above T-s diagram it can be easily seen that the coloured area is the increase in net work output on account of lowering the condenser pressure from P_4 to P_4' .

Power loss due to excess back pressure



The energy balance also reveals that **two thirds of the fuel energy is lost in the condenser** and carried out into the environment, In addition, losses of energy can be large quantity while it is thermodynamically insignificant due to its low quality. Exergy- based efficiencies and losses, however, provide measures of approach to ideality or deviation from ideality.

7.7 HOW TO INCREASE EFFICIENCY OF A CONDENSER

The condenser efficiency can be increased by using “PINCH TECHNOLOGY”

Minimum temperature approach between two curves on a vertical axis i.e., ΔT is minimum.

Steam & cooling water requirements are reduced by same amount. Amount of heat recovery depends upon “Minimum approach” temperature allowed for heat exchanger

Pinch Technology provides a systematic methodology for energy saving in processes and total sites. The methodology is based on thermodynamic principles. Figure 1 illustrates the role of Pinch Technology in the overall process design.

Pinch technology: systematic methodology for energy saving in processes and total sites

- Energy target
- Cross pinch
- Heat exchanger network design

Energy Targets

Starting from the thermal data for a process (such as shown in Table 1), Pinch Analysis provides a *target* for the minimum energy consumption. The energy targets are obtained using a tool called the “Composite Curves”.

Determining the Energy Targets

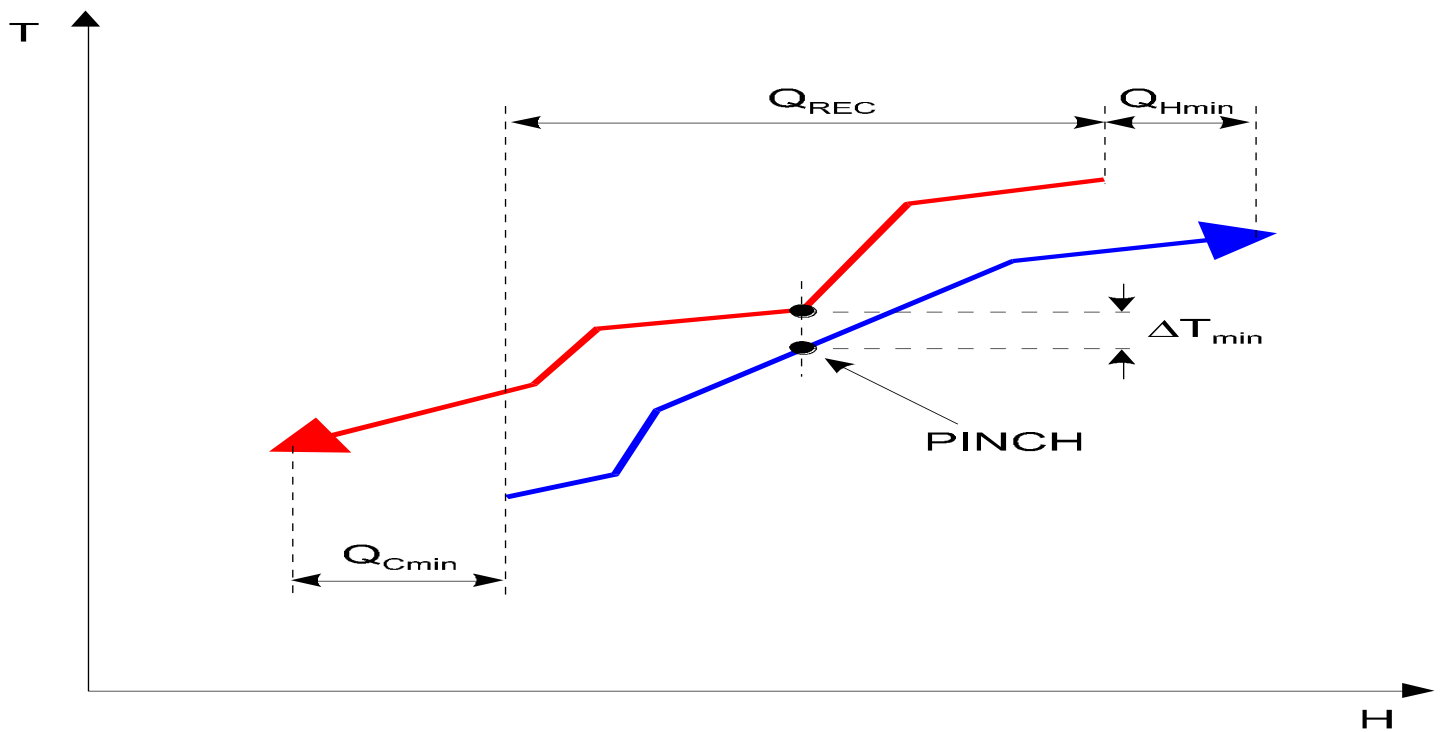
The composite curves provide a counter-current picture of heat transfer and can be used to indicate the minimum energy target for the process. This is achieved by overlapping the hot and cold composite curves, as shown in Figure, separating them by the minimum temperature difference DT_{min} ($10^{\circ}C$ for the example process). This overlap shows the maximum process heat recovery possible, indicating that the remaining heating and cooling needs are the minimum hot utility requirement ($Q_{H\ min}$) and the minimum cold utility requirement ($Q_{C\ min}$) of the process for the chosen DT_{min} .

7.8 Construction of Composite Curves

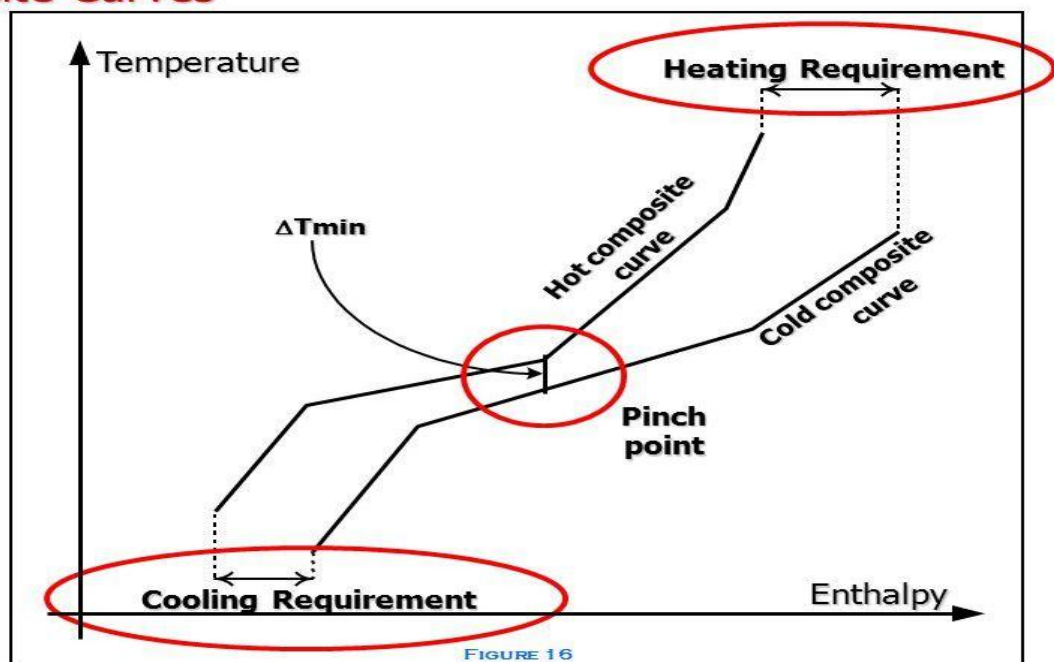
Composite Curves consist of temperature-enthalpy (T-H) profiles of heat availability in the process (the “hot composite curve”) and heat demands in the process (the “cold composite curve”) together in a graphical representation. Figure 3 illustrates the construction of the “hot composite curve” for the example process, which has two hot streams (stream number 1 and 2, see Table 1). Their T-H representation is shown in Figure 3(a) and their composite representation is shown in Figure 3(b). Stream 1 has a CP of $20\ kW/^{\circ}C$, and is cooled from $180^{\circ}C$ to $80^{\circ}C$, releasing $2000\ kW$ of heat. Stream 2 is cooled from $130^{\circ}C$ to $40^{\circ}C$ and with a CP of $40\ kW/^{\circ}C$ and loses $3600\ kW$.

Retrofit

Pinch Technology is applicable to both new design and retrofit situations. The number of retrofit applications is much higher than the number of new design applications. In this section techniques are discussed for setting targets for energy saving for an existing plant based on capital-energy trade-off for retrofit projects. The Super Target Process module developed by Linnhoff contains tools which employ these techniques.

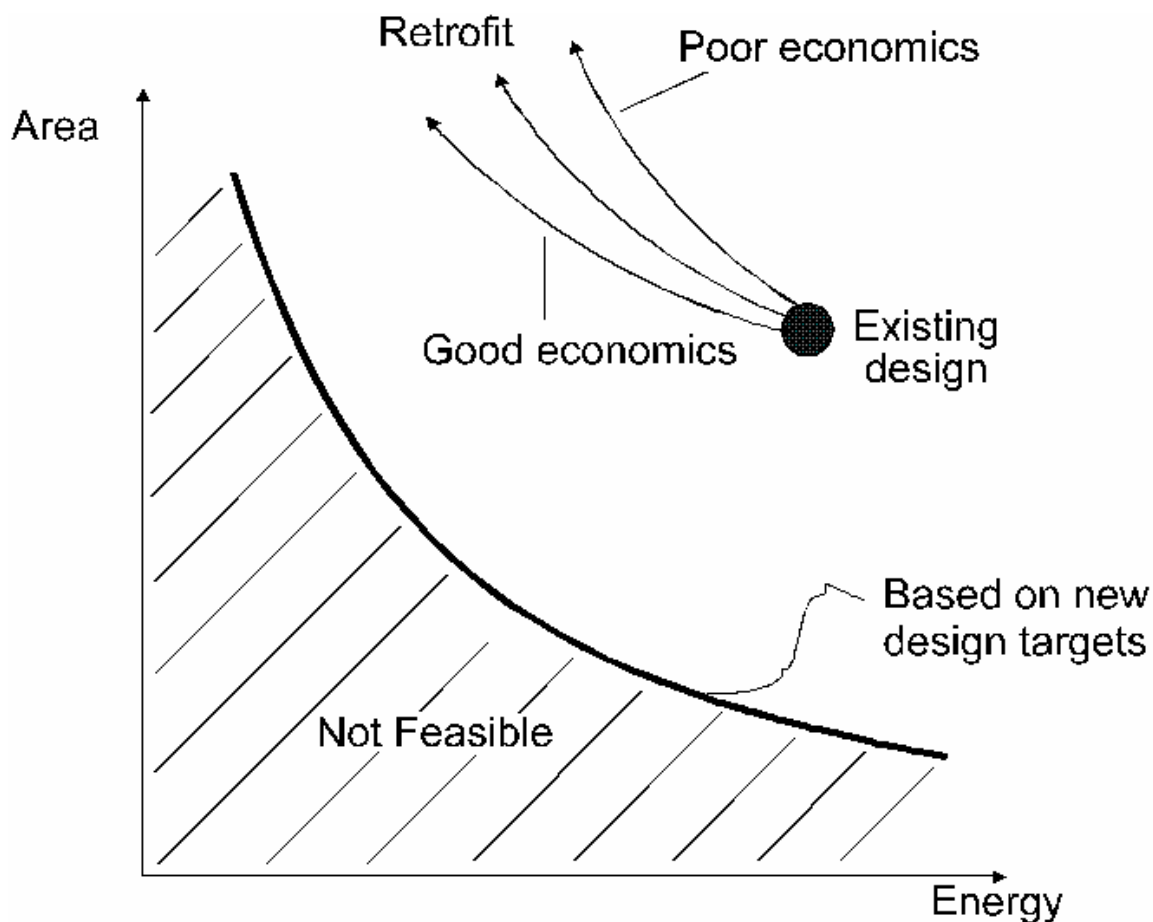


Composite Curves



7.9 Retrofit Targeting based on Capital-energy trade-off

Figure provides an understanding of the capital - energy trade-off for a retrofit project using an area-energy plot.



The curve (enclosing the shaded area) is based on new design targets for the process. The shaded area indicates performance better than the new design targets (which is infeasible for an existing plant). An existing plant will typically be located above the new design curve. The closer the existing plant is to the new design curve the better the current performance. In a retrofit modification, for increased energy saving, the installation of additional heat exchanger surface area is expected. The curve for the additional surface area that is closest to the new design area energy curve provides the most efficient route for investment (good economics).

8. Conclusion

The comprehensive thermodynamic modeling and multi-objective optimization of a combined cycle power plant with different vapor quality at steam turbine outlet provides good information. A complete thermodynamic modeling for this CCPP is performed. Exergy, exergo economic and exergo environmental of case also are conducted. In optimization process three cases with different quality at steam turbine outlet are analyzed and compared. Results show the system with 88% quality at steam turbine outlet is the most realistic in respect to efficient, economic and environmental aspects. Other significant conclusions are as follow:

- By increasing condenser efficiency by drawing composite curves using pinch technology.
- By changing the temperature & pressure parameters of steam turbine outlet quality.
- By increasing steam turbine outlet vapor quality, the temperature and pressure at steam turbine inlet flow will be increased.
- An increase in GTIT and also combustor working temperature result in an increase in system exergy efficiency.
- By applying the cost constraint reduces the overall cost of the cycle while increases the steam cycle quality.
- An increase in mass flow rate of HP steam and decreases LP steam mass flow rate result in an increase the exergy efficiency of the bottoming cycle.

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Streamlining and the impact exit quality of the steam turbine yield force of a joined cycle force plant

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Abstract

A restricted way exists to a maintainable arrangement which goes through watchful strides of efficiency change (asset administration) and gives natural amicable energies. Warm power plants are more regular in numerous force generation destinations around the globe. Consequently, in this ebb and flow examination concentrate on a thorough thermodynamic demonstrating of a consolidated cycle force plant with double weight heat recuperation steam generator is displayed. Since the steam turbine outlet quality is prohibitive parameters, advancement of three cases with distinctive steam quality are led and talked about. In other hand, vitality and exergy investigation of every segments for these three distinct cases evaluated and thought about. Gotten results demonstrate that it is truly essential to keep the nature of the vapor at turbine outlet steady in 88% for the outcomes to be more reasonable furthermore advancement and information are all the more in fact achievable and appropriate.

Keywords: cycle, exergy, investigation, quality, vapor.

INTRODUCTION TO COMBINED CYCLE

In electric power generation a combined cycle is an assembly of heat engines that work in tandem from the same source of heat, converting it into mechanical energy, which in turn usually drives electrical generators. After the cycle completion the working fluid of the first heat engine is still low enough in its entropy that a second subsequent heat engine may extract energy from the waste heat (energy) of the working fluid of the first engine. Possibly an overall efficiency of 51% (in a mechanical combination of two (2) cycles) in net Carnot thermodynamic efficiency. This can be done because heat engines are only able to use a portion of the energy their fuel generates (usually less than 50%). In an ordinary (non combined cycle) heat engine the remaining heat (e.g., hot exhaust fumes) from combustion is generally wasted.

Combining two or more thermodynamic cycles results in improved overall efficiency, reducing fuel costs. In stationary power plants, a widely used combination is a gas turbine (operating by the Brayton cycle) burning natural gas or synthesis gas from coal, whose hot exhaust powers a steam power plant (operating by the Rankine cycle). This is called a Combined Cycle Gas Turbine (CCGT) plant, and can achieve a best-of-class real (HHV - see below) thermal efficiency of around 54% in base-load operation, in contrast to a single cycle steam power plant which is limited to efficiencies of around 35-42%. Many new gas power plants in North America and Europe are of this type. Such an arrangement is also used for marine propulsion, and is called a *combined gas and steam (COGAS)* plant. Multiple stage turbine or steam cycles are also common.

Other historically successful combined cycles have used hot cycles with mercury vapor turbines, magnetohydrodynamic generators or molten carbonate fuel cells, with steam plants for the low temperature "bottoming" cycle. Bottoming cycles operating from a steam condenser's heat exhaust are theoretically possible, but uneconomical because of the very large, expensive equipment needed to extract energy from the small temperature differences between condensing steam and outside air or water. However, it is common in cold climates (such as Finland) to drive community heating systems from a power plant's condenser heat. Such cogeneration systems can yield theoretical efficiencies above 95%

3.4 Exergoenvironmental analysis

An important measure for reducing environmental impact, including emissions of carbon dioxide, a primary greenhouse gas, is increasing efficiency and thereby decreasing fuel use. Based on literature many studies are involved exergy and exergoeconomic analyses for CCPP, but many do not consider the environmental impact in their studies. Addressing this deficiency is one objective of this article, in which emissions of CO₂, CO

and NO_x are considered. NO_x formation and CO emission are more accurately related to the primary combustion parameters. Therefore, adiabatic flame temperature and also residence time in this region affect the NO_x and CO emission. High residence time increases NO_x but reduces the CO value. Thus, the analytical prediction of gas turbine combustion emission is provided. In this regard, the first step is definition on important parameters like adiabatic flame temperature. The adiabatic flame temperature in the initial zone of the combustion chamber is determined as follows [25,33]:

$$T_{pe} = A\sigma^x \exp[\beta(\sigma + \lambda)^2] \pi^x \theta^y \psi^z \quad (27)$$

where π is dimensionless pressure (P/P_{ref}), θ is dimensionless temperature (T/T_{ref}), and ψ is the H/C atomic ratio, $\sigma = \phi$ for $\phi \leq 1$ (ϕ is mass or molar ratio) and $\sigma = \phi - 0.7$ for $\phi \geq 1$. Moreover, x , y and z are quadratic functions of σ based on the following equations:

$$x^* = a_1 + b_1\sigma + c_1\sigma^2 \quad (28)$$

$$y^* = a_2 + b_2\sigma + c_2\sigma^2 \quad (29)$$

$$z^* = a_3 + b_3\sigma + c_3\sigma^2 \quad (30)$$

where A , α , β , λ , a_i , b_i and c_i are constants. Values for these parameters are listed in Table 1.

The amount of CO and NO_x produced in a combustion chamber and combustion reaction is dependent on the adiabatic flame temperature. The pollutant emissions (in grams per kilogram of fuel) can be identified as follows [34]:

$$\begin{aligned} m_{\text{NO}_x} &= \frac{0.15 \times 10^{16} \tau^{0.5} \exp(-71100/T_{pz})}{P_3^{0.05} (\Delta P_3/P_3)} \\ m_{\text{CO}} &= \frac{0.179 \times 10^9 \exp(7800/T_{pz})}{P_3^2 \tau (\Delta P_3/P_3)} \end{aligned} \quad (31)$$

where τ is the residence time in the combustion zone (assumed constant and equal to 0.002 s) [2], T_{pz} is the primary zone at combustion, P_3 is the combustor inlet pressure, and $\Delta P_3 = P_3$ is the non-dimensional pressure drop in the combustion chamber.

To calculate the value of CO₂ emission, it is better to define the net emission per kW h produced energy of the plant and try to reduce it [25]. This definition is a measure of environmental efficiency of the plant. This value talks about, how efficient the power plant is in terms of green house gases (GHG) emission. The value of CO₂ emission in combustion chamber can be calculated by using the following

simple equation:

$$\dot{m}_{\text{CO}} = 44.01 x \left(\frac{\dot{m}_{\text{fuel}}}{m_{\text{fuel}}} \right)$$

where x is the mole fraction of carbon in the fuel and m_{fuel} is the molar weight of the fuel. This simple equation estimates the CO₂

Table 1
Values for parameters in Eqs. (27)–(30)

Parameter	$0.3 \leq \phi \leq 1.0$		$1.0 \leq \phi \leq 1.6$	
	$0.92 \leq \theta \leq 2$	$2 \leq \theta \leq 3.2$	$0.92 \leq \theta \leq 2$	$2 \leq \theta \leq 3.2$
A	2361.76	2315.75	916.82	1246.17
α	0.1157	-0.049	0.288	0.381
β	-0.948	-1.114	0.145	0.347
λ	-1.097	-1.180	-3.277	-2.0365
a_1	0.014	0.010	0.031	0.036
b_1	-0.055	-0.045	-0.078	-0.085
c_1	0.052	0.0482	0.0497	0.0517
a_2	0.395	0.568	0.025	0.009
b_2	-0.441	-0.550	0.260	0.502
c_2	0.141	0.132	-0.131	-0.247
a_3	0.005	0.011	0.004	0.017
b_3	-0.129	-0.129	-0.178	-0.189
c_3	0.082	0.084	0.098	0.1037

emission accurately for complete combustion. In this case the total carbon dioxide is the sum of carbon dioxide produced in duct burner and gas turbine combustion.

Optimization

A multi-objective optimization method is applied to find the best design parameters of this CCPP. Objective functions, decision variables and constraints of this case are represented in this section

4.1.2. Total cost rate

By multiplying the respective flow rates of CO, NO_x and CO₂ emission in their corresponding unit damage cost (Table 2), the environmental impact is known as the total pollution damage (\$/s) in the second objective function. Therefore, in this research the cost of pollution damage is directly added to the expenditures that must be paid.

$$\dot{C}_{Total} = \dot{C}_f + \sum_k \dot{Z}_k + \dot{C}_{Env} + \dot{C}_D \quad (34)$$

where

$$\dot{C}_{env} = c_{CO} \dot{m}_{CO} + c_{NO_x} \dot{m}_{NO_x} + c_{CO_2} \dot{m}_{CO_2} \quad (35)$$

Table 2
Environmental damage costs [28].

Emissions	CO	NO _x	CO ₂
Cost (\$/Kg)	6.242	8.175	0.024

$$\dot{C}_f = c_f \dot{m}_f \times LHV \quad (36)$$

$$\dot{C}_D = c_f \dot{E}x_D \quad (37)$$

In which $\dot{Z}_k$ is the purchase cost of each component and $\dot{C}_f$, $\dot{C}_D$ are fuel cost and cost of exergy destruction, respectively.

4.1.3. CO₂ emission

Since the importance of environmental effects, CO₂ emission is considered as the third objective function which is produced in the combustion chamber and duct burner. Therefore, the CO₂ emission of the plant can write as follow.

$$\dot{C} = \frac{\dot{m}_{CO_2}}{\dot{W}_{net}} \quad (38)$$

Physical constraints

There are some physical constraints that must also be taken into account in cycle design. For instance, condensate pump pressure outlet must be less than feed pump first stage pressure outlet. Turbine listed in Table 3.

Decision variables

The decision variables are selected as 15 variables which define the thermodynamic of the cycle. Although the decision variables may be varied in the optimization procedure, each is normally required to be within a reasonable range. Such constraints, for these decision variables are listed in Table 4.

Evolutionary algorithm: genetic algorithm

There are lots of optimization techniques, among them evolutionary algorithm (EA) is known as one the important optimization techniques. They are a set of modern meta-heuristics used successfully in many applications with great complexity. Therefore, EAs are success on solving difficult problems [35,36].

RESULTS AND DISCUSSION

Optimization result and the effect of steam turbine outlet vapor fraction

Each optimization process has some constraints and the constraints in this case are the upper and lower bounds along with several physical criteria. Besides these constraints, the steam

Table 3
The list of constraints for optimization [4].

Constraints	Rationale
$T_{IT} < 1500 \text{ K}$	Material Temperature Limit
$r_{comp} < 0.9$	Commercial Availability
$\eta_{comp} < 0.9$	Commercial Availability
$\eta_{GT} < 0.9$	Commercial Availability
$P_{turb} < 110 \text{ bar}$	Commercial Availability
$0.75 < \eta_{ST} < 0.9$	Commercial Availability
$0.75 < \eta_{pump} < 0.9$	Commercial Availability
$m_{DH} < 2 \text{ kg/s}$	Super Heater Temperature Limitation
$5 \text{ bar} < P_{cond} < 15 \text{ bar}$	Thermal Efficiency Limitation
$T_{turb} < 550 \text{ }^\circ\text{C}$	Material Temperature Limitation
$120 \text{ }^\circ\text{C} \leq T_{13}$	To avoid formation of sulfuric acid in exhaust gases
$5 \text{ }^\circ\text{C} < PP < 30 \text{ }^\circ\text{C}$	Second Law of Thermodynamic Limitation

Table 4
List of decision variables with the upper and lower bounds [4].

Decision variables	Lower bound	Upper bound
Compressor pressure ratio	3	20
Compressor isentropic efficiency	0.75	0.94
Combustor outlet temperature (K)	1000	1500
Turbine efficiency	0.75	0.94
Duct burner fuel flow (kg/s)	0	1
HP Pinch point	1	8
LP Pinch point	10	32
Lower pressure steam mass flow (kg/s)	5	15
High pressure steam mass flow (kg/s)	50	100
Feed pump first stage pressure (bar)	5	15
Feed pump second stage pressure (bar)	90	120
Condensate pump pressure (bar)	2	8
Pump efficiency	0.6	0.9
Steam turbine efficiency	0.75	0.94
Condenser pressure (bar)	0.05	0.15

turbine outlet vapor quality is another really important parameter which greatly affects the optimization results. In the current design point, quality of the vapor at the turbine outlet is about 88%. It is really important to keep the quality of the vapor constant for the results to be more realistic and also optimization and data are more technically feasible and applicable.

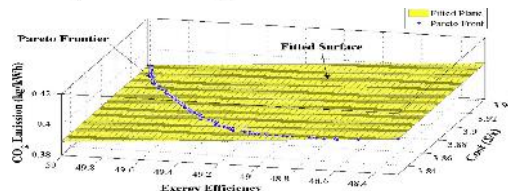


Fig. 3. 3D frontier and plan of feasible solutions for the case of 80% vapor quality at steam turbine outlet.

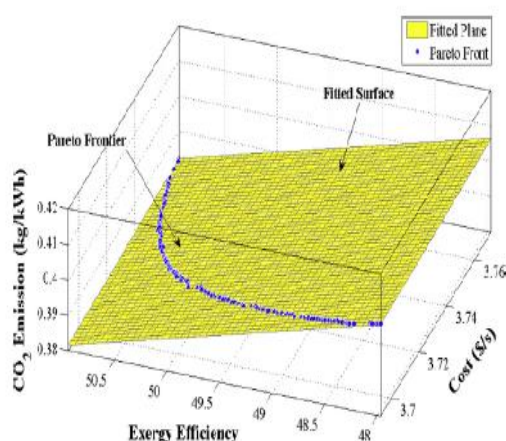


Fig. 2. 3D frontier and plan of feasible solutions for the case of no constraint on the quality.

Pareto frontier and selection of best point

Pareto frontier contains best set of solution family of the problem. One of the asymptotic surface of the Pareto frontier. A solution which is closer to this point is referred to be the best solution. Since the data lies on a 3D plan, one way to reduce the complexity of finding the equilibrium point is by transforming the coordinates and the Pareto curve to the corresponding fitted plan coordinates. Through this method, the problem changes to a 2D problem and after the optimum point is found, data are transformed back to the original 3D plan.

As discussed, the data from the 3D space was transformed into the 2D plan coordinates and the third coordinate was eliminated. The modified cost and efficiency

values were plotted and the equilibrium and the best point solutions are calculated and selected. Figs. 5–7 show the procedure of selecting the best solution for

the three mentioned cases. In this respect, a set of asymptotic are plotted for each Pareto frontier and their intersection shows the equilibrium line. Thus, the best solution is the point which has the minimum distance from the equilibrium point. On the other side, the optimum point seems to be farther than some other points on the curve due to the coordinates are not in the same scales. In addition, the procedure of calculating the equilibrium point and minimum distance point is automatically done in the programme and are applied to all cases.

Cost constraint

Equilibrium point method leads to the following tabulated performance data for each case as the best case. By considering cost limits and efficiency variations, one may conclude that higher vapor quality at the turbine outlet means higher cost, lower efficiency and also lead to the higher cost variation rate per efficiency improvement. In fact, high vapor quality at the outlet requires employment of better technology in the plant which could not be cost effective. The data of each case as the best case that was calculated using the equilibrium point method is shown in Table 6. By comparing the data calculated from the equilibrium point method with the current design data, it is observed that the cost of third case for its optimal solution is much higher than the original design point. Therefore, this case cannot be a solution due to the cost of employing this case is much higher than the accepted value. On the other hand, if the financial resources are limited to a specific value the method of equilibrium point must be used with a cost constraint. In this case the cost constraints depend on the current cost of the plan. The process of best point selection is shown in Fig. 8.

Table 7
Best point using equilibrium point and cost constraint approach for case 3.

	Cost (\$/s)	Efficiency	CO ₂ (kg/kWh)
3 (quality = 0.88) (equilibrium method)	6.325	0.482	0.404
3 (quality = 0.88) (cost constraint)	4.412	0.475	0.410
Design point	4.45	0.448	0.435

.. essure and low temperature are needed in HRSG. Consequence, lower temperature and pressure of the cycle increase as illustrated in Fig. 9. In Fig. 10 the effects of quality control on the optimum gas cycle are presented. To produce a higher temperature steam, the turbine inlet and outlet temperatures need to be high enough. An increase in gas turbine inlet temperature and also combustion chamber working temperature, results in an increase in the cost

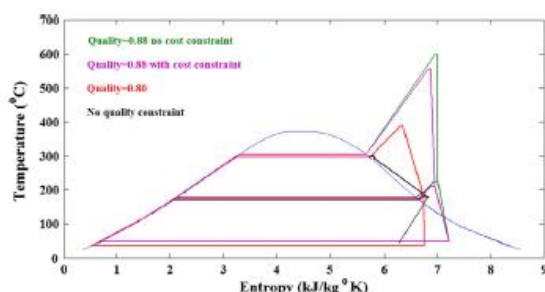


Fig. 9. T-S diagram of steam cycle for different cases.

of components. When cost constraint is applied, the cycle performance changes significantly. Lower isentropic efficiency of the compressor leads to a lower compressor cost. Moreover, by applying cost constraint, a fuel flow reduction occurred in the duct burner which leads to reduce the fuel cost, HRSG cost, duct burner cost and also the cost of steam cycle. Thus, cost constraint reduces the overall cost significantly.

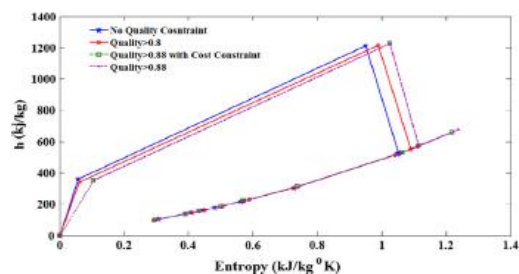


Fig. 10. h-s diagram of gas cycles for different cases.

Furthermore, the components efficiency of the gas cycle will be lower and entropy changes in the gas cycle are higher. Fig. 11 provides a comparative presentation for the decision variables. In this graph, each decision variable is

normalized by its maximum value for 4 cases. As it is obvious in this figure, for the steam quality outlet of 88%, the compressor efficiency and compressor pressure ratio are significantly lower which causes lower cost and higher gas cycle outlet temperature. Steam turbine efficiency has the highest value for the quality of 88% outlet with no cost constraint, efficiency to be selected regardless of the quality constraint at outlet. Duct burner fuel mass flow rate is the lowest for the case of no constraint due to the fact that, chemical reactions are the most destructive phenomenon. An increase in fuel mass flow rate causes extra cost flow and extra emission in addition to the gas cycle. So, if there is no constraint, less fuel will be burnt in the duct burner which decreases the cycle efficiency. However, the exergy of exhaust gases extracted only to produce more work. Therefore, this system with 88% quality at steam turbine outlet and cost constraint is the most efficient, economical and environmental friendly.

Mass flow rate of HP steam increases when the steam quality reaches the higher values but LP steam mass flow rate is reduced. In the same manner, pinch point temperature of LP drum increases while HP drum pinch point reduces. In general when quality is under control and needs to be high, the optimization tool tries to find cycles with higher bottoming cycle efficiency.

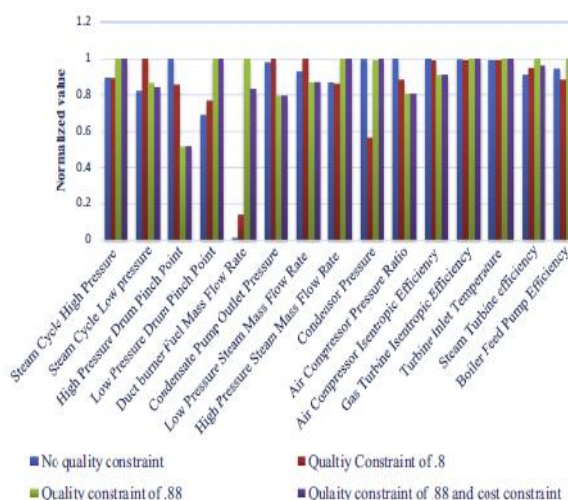
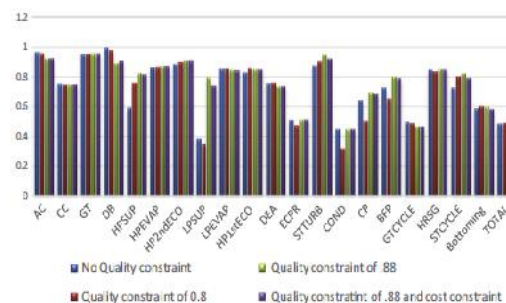
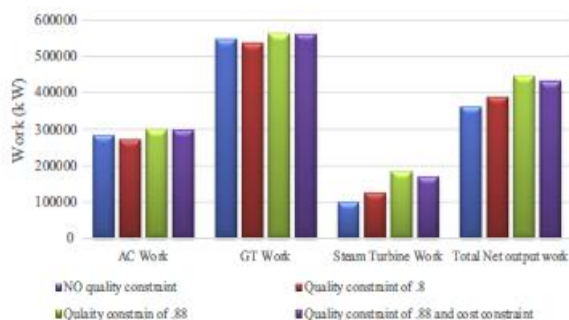
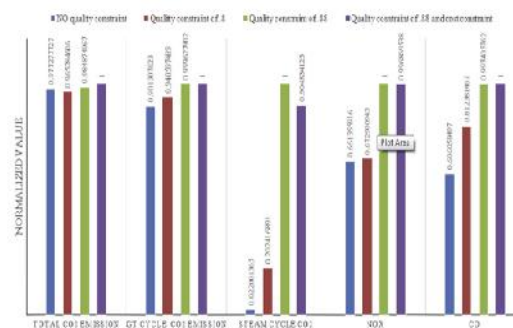


Fig. 11. Comparative presentation of decision variables for different case.



The purchase costs of the power plant components are pre-sented in Fig. 15. They are normalized with the maximum value in three cases with different qualities and also one case with cost constraint, this figure provides a comparative method of system



In this section, the basic design data is compared with the optimum cycle data which was concluded from the optimization results. Therefore, the data of decision variables for basic design and optimum point are listed in Table 8. In order to compare the cycles in an easy manner, variation of the design variables from actual data is shown in Fig. 17. As it is clear in this figure, gas turbine cycle parameters increase except compressor efficiency. Steam cycle mass flow rate decreases especially for low pressure mass flow rate while the efficiencies of the

components in bottoming cycle are increased.

Table 8
Decision variables for design and optimum cycles [1].

Decision variable	Optimized	Design	Unit
Steam Cycle High Pressure	103.542	110.460	Bar
Steam Cycle Low Pressure	9.012	11.280	Bar
High Pressure Drum Pinch Point	2.537	4.000	K
Low Pressure Drum Pinch Point	22.800	24.500	K
Duct burner Fuel Mass Flow Rate	0.804	0.800	Kg/s
Condensate Pump Outlet Pressure	2.497	5.000	Bar
Low Pressure Steam Mass Flow Rate	12.904	18.000	Kg/s
High Pressure Steam Mass Flow Rate	132.347	134.200	Kg/s
Condenser Pressure	0.112	0.090	Bar
Air Compressor Pressure Ratio	11.066	10.100	(-)
Air Compressor Isentropic Efficiency	0.808	0.864	(-)
Gas Turbine Isentropic Efficiency	0.903	0.881	(-)
Turbine Inlet Temperature	1336.780	1244.150	K
Steam Turbine Efficiency	0.898	0.840	(-)
Boiler Feed Pump Efficiency	0.872	0.750	(-)

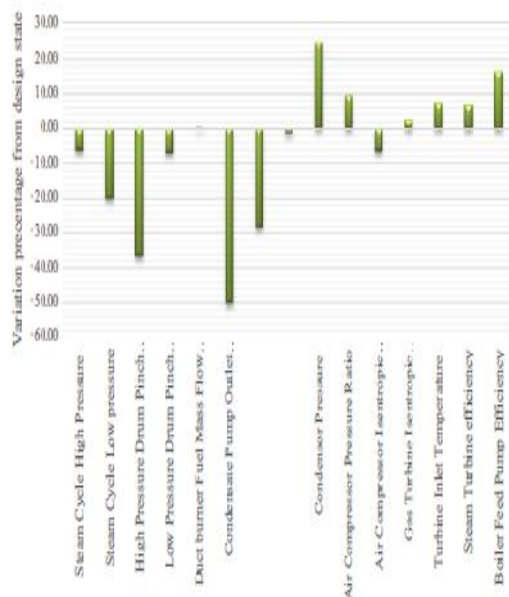


Fig. 17. Changes in variables after optimization.

CONCLUSION

The comprehensive thermodynamic modeling and multi-objective optimization of a combined cycle power plant with different vapor quality at steam turbine outlet provides good information. A complete thermodynamic modeling for this CCPP is performed. Exergy, exergoeconomic and exergoenvironmental of case also are conducted. In optimization process three cases with different quality at steam turbine outlet are analyzed and compared. Results show the system with 88% quality at steam turbine outlet is the most realistic in respect to efficient, economic and environmental aspects. Other significant conclusions are as follow:

- By increasing steam turbine outlet vapor quality, the temperature and pressure at steam turbine inlet flow will be increased.
- An increase in HRSG high pressure and decrease HRSG low pressure, increases system exergy efficiency and decreases the total rate of the system.
- An increase in GTIT and also combustor working temperature result in an increase in system exergy efficiency an increase in total cost of the system.
- By applying the cost constraint reduces the overall cost of the cycle while increases the steam cycle quality.

An increase in mass flow rate of HP steam and decreases LP steam mass flow rate result in an increase the exergy efficiency of the bottoming cycle.

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