Gas Turbine/HRSG Co-generation Cycle Performance Evaluation and Enhancement

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This thesis is submitted as a partial fulfillment of the requirements for the Degree of Bachelors of Engineering (BE) in Chemical Engineering

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Dedication

We dedicate this thesis to our
parents, family, friends and teachers

And above all,

To Allah Almighty.

CERTIFICATE

This is to certify that work in this thesis has been carried out by **Mr. Sheikh Ehsan-ul-Haq, Ms. Mashal Zahid** and **Mr. Bilal Khalid** and has been completed under the supervision of **Dr. Bilal Khan Niazi** and **Dr. Muhammad Ahsan** at the School of Chemical and Materials Engineering (SCME), National University of Science and Technology, H-12, Islamabad, Pakistan.

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Abstract

The project has two major units in it, namely ‘Gas Turbine’ & ‘HRSG’. The former utilizes natural gas, post combustion as is fuel, which in turn rotates generator having a capacity of 9MW, whereas the latter employs the exhaust gases coming out from the gas turbine to generate the steam needed in the process called ‘steam reforming’.

Our aim is to enhance the efficiency of this whole unit, in order to do so, we will lower the temperature of exhaust gases via adding an additional unit to exchange heat with flue gases, as there is still some potential in the gases going into the atmosphere. Doing so will not only enhance the efficiency of the whole process but also will lower the cost per unit of electricity generated.

Nomenclature

ΔP_c	kPa or bar	pressure drop due to catalyst bed
V_p	V^3	volume of particle
ΔPb_1	kPa or bar	pressure drop due to inert alumina balls on top
ΔPb_2	kPa or bar	pressure drop due to inert alumina balls at the bottom
ΔP_t	kPa or bar	total pressure drop of catalyst bed and alumina balls
T_{in}	K or ℃	temperature of feed stream at reactor inlet
T_{out}	K or ℃	temperature of product stream at reactor outlet
$-\Delta H_{rxn1}$	kJ/kmole	heat of reaction for reaction 1
$-\Delta H_{rxn2}$	kJ/kmole	heat of reaction for reaction 2
ρm	kmol/m^3	molar density
ΔT	K or ℃	temperature difference
C_p	kJ/kmol℃	heat capacity
D_i	m	internal diameter of shell
S	bar	maximum allowable stress at design temperature property of material of construction
E	-	weld efficiency
P_d	bar	design pressure of vessel
t_a	m	shell thickness

t_b	m	hemispherical head thickness top and bottom
m'	kmol/hr	molar flow rate
C_c	kJ/hr.K	product of molar flow rate and heat capacity for cold stream
C_h	kJ/hr.K	product of molar flow rate and heat capacity for hot stream
Ɛ	-	heat transfer effectiveness
NTU	-	number of transfer units
U	$W/m^2.K$	overall heat transfer coefficient
A_s	m^2	required surface area
L_{eff}	m	effective length of each tube
d_o	m	outer diameter of tube
d_i	m	inner diameter of tube
V''	m^3/s	volumetric flow rate
A_{st}	m^2	surface area of each tube
N	-	number of tubes
j_f	-	friction factor
N_p	-	number of passes

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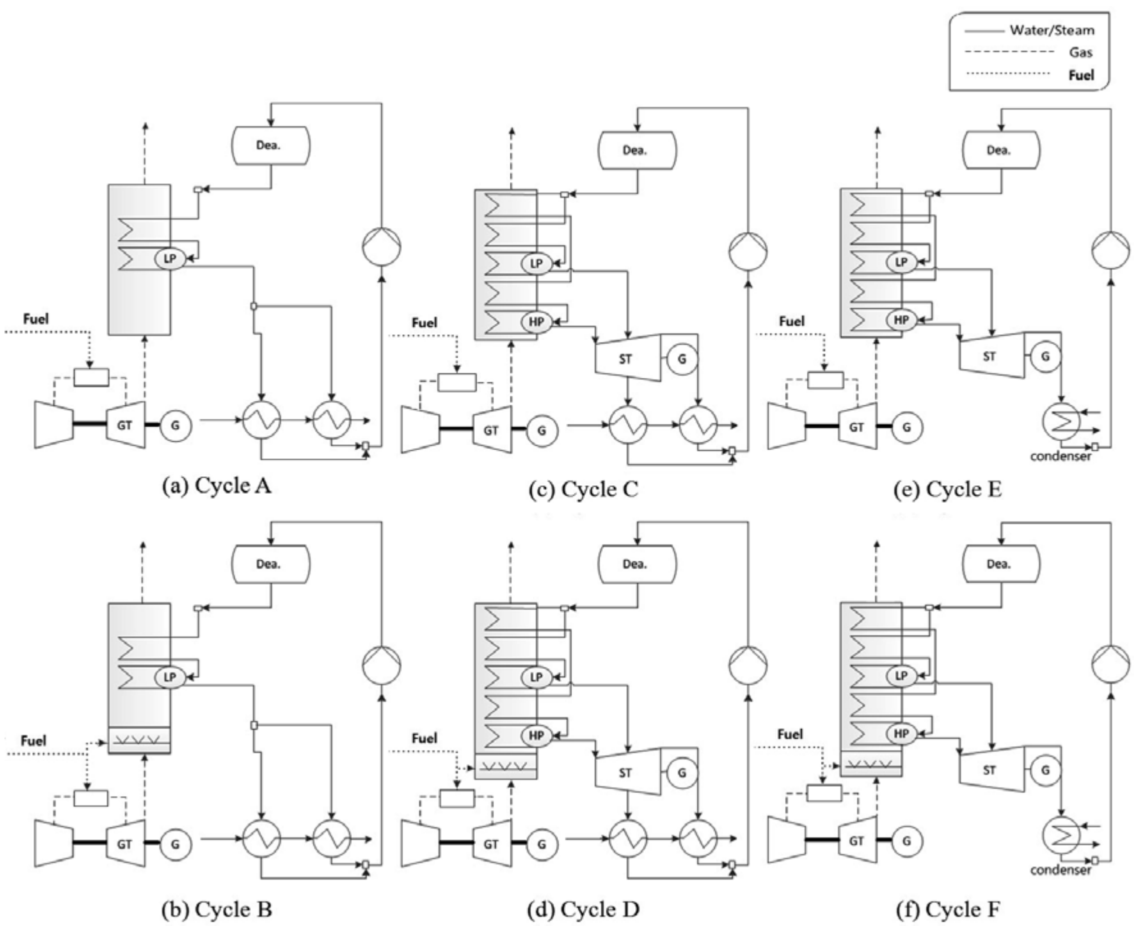
CHAPTER–1: Literature Reviews of Related Topics' Papers

1.1 - Performance Evaluation of Combined Heat and Power Plant Configurations

It has been the desire of all the engineers around the globe to create power plants with efficiencies approaching ideality, as we know, it is the process of evolution and day by day with the advancement of technology the efficiency is improving. However, the pinnacle is yet to come. This research is also the part of the progress taking place. The paper is focusing on different calculations that can evaluate and compare the performance of combined cycle power plants being used for specific applications, by taking into account their inputs, outputs and finally the cost of production of per kWh of energy. The techniques include electrical efficiency, utilization



(a) Cycle A (c) Cycle C (e) Cycle E

(b) Cycle B (d) Cycle D (f) Cycle F

factor and R1-energy efficiency. The study considers three different types of combined cycle power plants, each of which is considered twice as with and without supplementary firing.

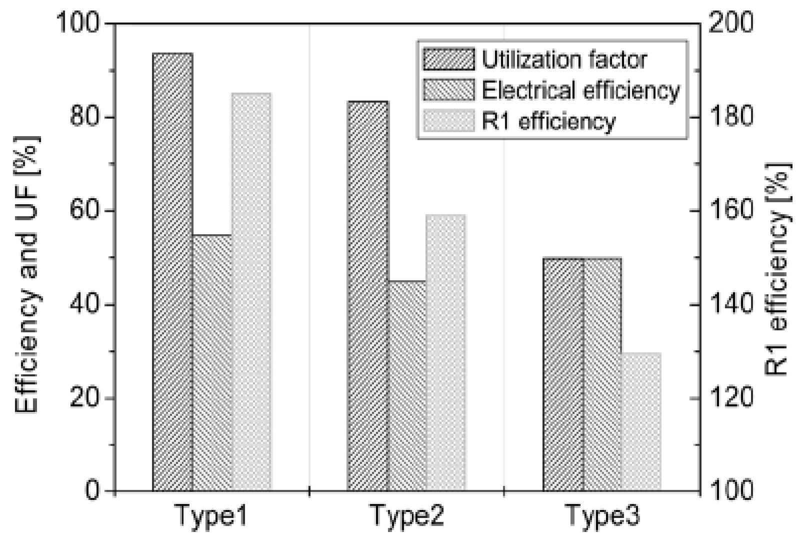
The first plant is producing electricity and heat for a desalination plant, the second plant under consideration is generating electricity and heat, the latter being only produced in winters for district heating. The third and the final type is producing only electricity.

Plant 2 has an ability to produce heat only in winter due to the availability of different operational modes. The first one is 'Back pressure mode' here the steam coming out of steam turbines is not condensed but is exchanged heat with the water which then goes to fulfil the needs of the district. The second mode of operation is 'condensing mode', as the steam which comes out of turbines is sent to a condenser where heat dissipates and the water made out of the steam is sent to the boiler again for heating and the cycle continues.

The evaluation of the CCPPs starts with the evaluation of their thermal efficiencies by considering the output and input energy. The other two efficiencies include utilization factor; this evaluates both the types energy produced by the CCPPs power and heat and the total energy given to the plant under study. The third technique being used is the most refined and sophisticated of the above. R1 efficiency has two equivalence factors; they are derived from the divided efficiency of electricity and heat generation power plants with electric power plants running at an average efficiency of 38%, inverse of it gives equivalence factor of 2.6 and heat plants have an average of 91% efficiency, so 1.1 comes as the equivalence factor.

The figures below have all the summary of calculations performed in this study. The major calculations were involved by using the stated formulas:

$$R1\,efficiency = \frac{2.6 \times W_{output} + 1.1 \times Q_{output}}{Q_{input}}$$



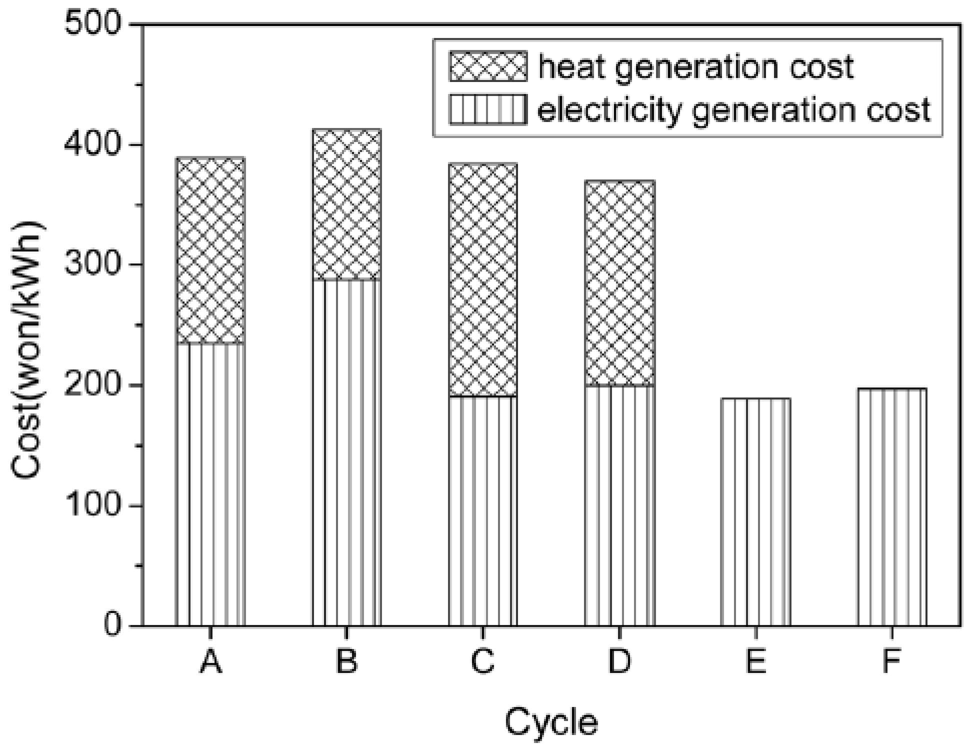
The first equation gives us the cost of per kWh of 'electricity' production, the second gives us the cost of per kilowatt-hour 'heat' production and the final equation is giving 'revenue' generated by selling electricity and heat.

$$Y_{El.yr} = \frac{I_c \times \Psi}{8760} + \frac{I_{fuel} \times C}{\eta_{elec}}$$

$$Y_{H.yr} = \frac{\Psi}{8760} \times \left(\frac{I}{\Phi} - I_c\right) + \frac{I_{fuel} \times C \times \Phi_{BR}}{\Phi}$$

$$Possible\ Revenue = \frac{\left(Y_{El.yr} + Y_{H.yr} \times \frac{\Phi}{\Phi_{El}}\right) \times 8760}{I_c + I_{fuel} \times 0.25 \times 8760}$$

In a nutshell, it is revealed that R1 equalized efficiency is the best parameter to compare the plants having different outputs of incomparable magnitude, as in this case plant 1 and 2 were producing heat and electricity, but the third one was not producing 'heat'. So, R1 enables us to compare divided generation with combined generation. The other efficiency methods cannot be considered for ultimate comparison among the plants for reliable results as they take into account only one of the product of the plant.



1.2 - Influence of ambient temperature on combined-cycle power-plant performance

Pakistan has been power stressed country since its inception; one solution might be the addition of turbines to power generation system. It will not only lessen the load on the hydroelectric power generation but will also give a sustainable solution to water fluctuation in dams thereby resulting in an unexpected shortage of power in the national grid. According to times of Islamabad, Tarbela dam power generation capacity reduced from 3,478 megawatts to only 297 megawatts, due to seasonal variations in January 2017. So the addition of such thermal power generation would stabilize the power output. In the period of last 2 to 3 years, the availability of natural gas, the primary fuel for Gas turbine, has improved to a sustainable extent. By 2018 it will be even better by the completion of multi-Billion Dollar project, namely, TAPI.

This paper will bring the variations in different parameters during the operation of turbines. The operation of combined cycle is quite sensitive to conditions at which the process is taking place, most notably atmospheric pressure, ambient temperature and relative humidity of the air. The temperature is the major focus of this discussion. However, minor details of the former two are also brought into the limelight.

The model used in this study has two gas turbines, 175MW each, with three stage steam turbines having the net power of 253MW and the net power is 600MW, with supplementary firing. To see the deviation we need standards so in this case, we have ISO conditions as a reference which is "Temperature: 15˚C, Pressure: 101.32 Pa, Relative Humidity: 0.6"

The brief explanation for the effect of the rise in temperature is that with the rise in temperature the specific mass of air reduces due to decrease in density, so the same volumetric flow has lesser specific mass which is being sent to the compressor and consequently lesser power generation. Coming to the effect of pressure, lesser pressure than atmospheric results in lower density and similar effects to that of temperature.

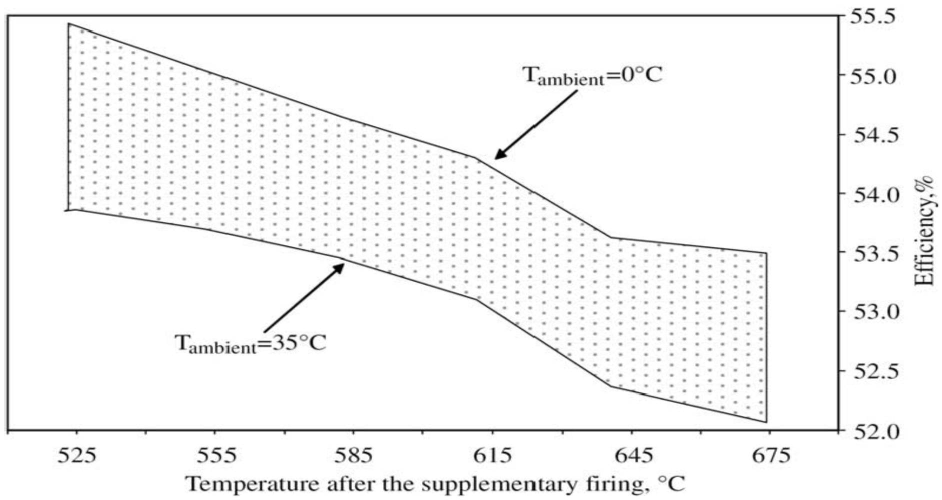
Finally, if the all other factors are kept constant, increasing relative humidity increases the power generated by the combined cycle. However, there is a slight downfall in power generated by the gas turbine, but the flue gases coming out of gas turbine are hotter than they were at ISO conditions. So overall there is a rise in power generation, this nevertheless is slight in magnitude.

Now let us analyse the effect of temperature in depth, but before we start one fact of core importance but be stated, which is "The temperature of gases posts complementary firing must not go beyond 675 ˚C, if it does steam formation will occur in economizer tubes." It has been concluded by the studies that keeping the ambient temperature undisturbed and varying the temperature of gases after supplementary firing can result in 77 MW of additional power from steam turbines.

Merely verifying ambient temperature results in 75 MW of power variation, taking the gas turbines from 380 MW at least ambient temperature to 305 MW at maximum ambient temperature. The thermal efficiency also gets affected by the ambient temperature and supplementary firing.

There had been a variation of 170 MW by changing the ambient temperature and temperature of gases going into steam turbines after firing. The minimum power generated was 468MW with highest ambient temperature and lowest temperature after supplementary firing, whereas a highest peak of 642 MW was seen at vice versa conditions.

The trends depict that that with lesser ambient temperature the efficiency is higher and with high temperature after supplementary firing reduces it. It can be visualized as follows:



To, reiterate in a nutshell, it can be seen that: ambient temperature has a notable effect upon the power generated by the gas turbines, the rise in temperature after supplementary heating reduces thermal efficiency. However, megawatts of power generated goes up. Using both factors of supplementary firing and ambient temperature side by side can result in 170 MW of additional power generation, with the thermal efficiency downfall of 3-4%.

1.3 - Boiler & HRSG performance evaluation

Thermal performance evaluation gives profitable data on a few parts of the boiler performance, distinguishes potential issue ranges and tackles numerous issues looked by the administrators of boilers and HRSGs. To guarantee that the model is sensible, the creator utilizes remedy factors in light of field information. Subsequently, if the plant has enough field information and the tube geometry information, the boiler/HRSG performance can be anticipated for any arrangement of conditions or varieties in sustain water temperature, steam pressure, vent gas stream, examination and so on.

One of the administrations is assessing the thermal outline and performance checking of any boiler or HRSG (new, to be purchased or existing). It is fitting to do this evaluation before the request for the boiler/hrsg is put. Genuine blunders if any can be amended early else the end client is screwed over thanks to the hardware for twenty to thirty years. It had been seen that boiler and HRSG providers don't give much data on the boiler thermal plan and performance viewpoints. What number of at last client's camp know whether the boiler dissemination is appropriate, regardless of whether the steam speed utilized as a part of the superheater tubes is proper considering the most extreme and least loads, is the balance thickness utilized proper for the superheater, does the surface territory mean much or what is the heater warm motion. Many end clients have no idea to assessing these focuses. Obviously they are not there to play out these sorts of studies. Henceforth anybody intending to put a huge number of dollars in another boiler/hrsg, have the thermal plan assessed by a free advisor who can propose upgrades to the plan before requesting the hardware. He can likewise acquire the information required for any future evaluation from the boiler provider. Numerous boiler providers don't give the fundamental information required to the thermal outline evaluation and consequently on account of any major issue at a later date, it ends up noticeably hard to do a boiler thermal performance investigation. On the off chance that the boiler/hrsg is as of now purchased and you have issues, at that point you will be moving toward the boiler provider in the event of any issue and most likely get a one-sided investigation which could possibly take care of the issue. On the off chance that you might want an autonomous expert to survey the plan, at that point the accompanying data must be given:

a) Performance of boiler/hrsg as recommended in the proposition. Information, for example, sustain water temperature, pressure of steam, steam stream, gas/steam temperature profiles at different warming surfaces ought to be accessible and vent gas stream and investigation (in the event of a waste warm boiler or HRSG). Likewise tube geometry information ought to be given. Numerous boiler providers don't give these information to their clients (and the client does not know what information to request) and accordingly, when some performance issue manifests, the boiler provider can rationalize and escape clean. This article will empower the potential boiler client to ask the correct inquiries to the boiler/Hrsg provider some time recently buying the unit.
b) Field information is an absolute necessity to guarantee that the model is dependable. Adjustment elements can be mimicked for each surface which can calibrate the model to coordinate the field

information and at that point one can check whether the first proposition was sensible. Say a HRSG is working at part stack. The thermal model can be reproduced utilizing the current field information from which one can extrapolate to some other arrangement of gas conditions and guarantee the first advertising was great.

Therefore, in this way some understanding of the basic boiler performance might be acquired which can offer assistance us to change the plan if totally required or see where the issues are. It can likewise tell if the first outline was traded off or satisfactory or not. This sort of examination is additionally helpful regardless of the possibility that the boiler is performing great. On the off chance that the plant needs to change the fuel utilized as a part of the boiler at a later date or change the channel gas stream, temperature or investigation to the HRSG, the boiler or HRSG performance ought to be assessed to guarantee no issues emerge. Changing to a low BTU fuel builds the vent gas amounts, and the gas pressure drop can influence the fan operation. One may likewise play out this evaluation to check whether there is the degree for enhancing the boiler productivity or vitality recuperation from the HRSG. In the event that the plant needs to build the limit of the boiler by 10%, one can instantly check if the pressure drops in the superheater are sensible and the drum pressure does not approach the planning pressure expecting tube divider temperatures are sensible .If the superheater is supplanted with containers of a diverse width sooner or later, by what method should the streams (number of tubes conveying the steam stream) be acclimated to guarantee sensible steam speed or pressure drop.

1.4 - Optimization of heat recovery steam generator through energy analysis for combined cycle gas turbine power plants

This paper proposes a new approach to finding the optimum design parameters of the heat recovery steam generator (H RSG) system to maximize the efficiency of the steam turbine (bottom) cycle of the combined cycle power plant (CCPP), but without performing the bottom cycle analysis. This could be achieved by minimizing the unavailable energy (the sum of the destroyed and the lost energies) resulted from the heat transfer process of the HRSG system. The present approach is relatively simple and straightforward because the process of the trial-and-error method, typical in performing the bottom cycle analysis for the system optimization, could be avoided. To demonstrate the usefulness of the present method, a single-stage HRSG system was chosen, and the optimum evaporation temperature was obtained corresponding to maximum useful work for given conditions of water and gas temperatures at the inlets of the HRSG system. Results show that the optimum evaporation temperature obtained based on the present exergy analysis appears similar to that based on the bottom cycle analysis. Also shown is the dependency of number of transfer unit (NTU) on the evaporation temperature, which is another important factor in determining the optimum condition when the construction cost is taken into account in addition to the operating cost. The present approach turned out to be a powerful tool for optimization of the single-stage. HRSG systems and can easily be extended to multi-stage systems. Another way of dealing with finding the optimum design parameters of the heat recovery steam generator (HRSG) framework, is proposed in this paper to expand the efficiency of the steam turbine (base) cycle of the combined cycle control plant (CCPP), but without playing out the base cycle examination. This could be accomplished by limiting the inaccessible energy (the total of the demolished and the lost energies) came about because of the heat exchange procedure of the HRSG framework.

The present approach is moderately basic and direct in light of the fact that the procedure of the experimentation technique, run of the mill in playing out the base cycle investigation for the framework optimization, could be maintained a strategic distance from. To, exhibit the handiness of the present strategy, a single-stage HRSG framework was picked, and the optimum dissipation temperature was achieved comparing to most extreme helpful work for given states of water and gas temperatures at the deltas of the HRSG framework. Results demonstrate that the optimum vanishing temperature acquired in view of the present vitality investigation seems like that in light of the base cycle examination. Additionally indicated is the reliance on a number of exchange unit (NTU) on the dissipation temperature, which is another vital factor in deciding the optimum condition when the development cost is considered notwithstanding the working expense.

The present approach ended up being an intense apparatus for optimization of the single-stage HRSG systems and can without much of a stretch be reached out to multi-stage systems.

1.5 - Economic & Technical Considerations for Combined-Cycle Performance-Enhancement Options

Output Enhancement

Plant output enhancements can be categorized further into two major categories: gas turbine inlet cooling and power augmentation.

Gas Turbine Inlet Air Cooling

For applications where significant power demand and highest electricity prices occur during the warm months, a gas turbine air inlet cooling system is a useful option for increasing output. Inlet air cooling increases output by taking advantage of the gas turbine's characteristic of higher mass flow rate and, thus, output as the compressor inlet temperature decreases. Industrial gas turbines that run at constant speed are constant-volume-flow machines. The specific volume of air is directly proportional to the temperature. Because the cooled air is denser, it gives the machine a higher air mass flow rate and pressure ratio, resulting in an increase in output. In combined-cycle applications, there is also a small improvement in thermal efficiency.

Evaporative Cooling Theory

Evaporative cooling works on the principle of reducing the temperature of an air stream through water evaporation. The process of converting the water from a liquid to a vapor state requires energy. This energy is drawn from the air stream. The result is cooler, more humid air. Theoretically, the minimum temperature that can be achieved by adding water to the air is equal to the ambient wet-bulb temperature. Practically, this level of cooling is difficult to achieve. The actual temperature drop realized is a function of both the equipment design and atmospheric conditions. Other factors being constant, the effectiveness of an evaporative cooling system depends on the surface area of water exposed to the air stream and the residence time. The effectiveness of the cooler is a function of its design and is defined as follows:

Cooler effectiveness = T_1 DB – T_2 T_1 DB – T_1WB

- 1 refers to entering conditions.
- 2 refers to exit conditions.
- DB equals dry-bulb temperature.
- WB equals wet-bulb temperature.

Typical effectiveness levels are 85 to 95%. Assuming the effectiveness is 85%, the temperature drop can be calculated by:

Temperature drop = 0.85 ($T_1DB - T_2WB$)

Foggers

Foggers were first applied to gas turbine inlet air cooling in the mid-1980s. Nearly 100 fog systems are installed on turbines in North America, from aero-derivatives to large-frame machines. Fog systems create a large evaporative surface area by atomizing the supply of water into billions of super-small spherical droplets. Droplet diameter plays an important role with respect to the surface area of water exposed to the airstream and, therefore, to the speed of evaporation. For instance, water atomized into 10-micron droplets yields 10 times more surface area than the same volume atomized into 100-micron droplets. For evaporative cooling or humidification with atomized water, it is important to make a true fog, not a mist. To a meteorologist, water droplets of less than 40 microns in diameter make up a fog. When droplet sizes are larger than this, they are called a mist. True fogs tend to remain airborne due to Brownian movement—the random collision of air molecules that slows the descent of the droplets—while mists tend to descend relatively quickly. In still air, for example, a 10-micron droplet falls at a rate of about one meter in five minutes, while a 100-micron droplet falls at the rate of about one meter in three seconds.

Power Augmentation

Three basic methods are available for power augmentation: water or steam injection, HRSG supplementary firing and peak firing.

Gas Turbine Steam/Water Injection

Injecting steam or water into the head end of the combustor for NO_x abatement increases mass flow and, therefore, output. Generally, the amount of water is limited to the amount required to meet the NO_x requirement to minimize operating cost and impact on inspection intervals. Steam injection for power augmentation has been an available option for over 30 years. When steam is injected for power augmentation, it can be introduced into the compressor discharge casing of the gas turbine as well as the combustor. In combined-cycle operation, the cycle heat rate increases with steam or water injection. In the case of water injection, this is primarily due to the use of high-grade fuel energy to vaporize and heat the water. In the case of steam injection, this is primarily due to the use of bottoming cycle energy to generate the steam for the gas turbine that could otherwise be used in the steam turbine. A secondary factor is that typical control systems reduce firing temperature when injecting steam or water. This counteracts the effect of higher heat transfer due to the extra water vapor on the gas side to maintain hot gas path part life. GE gas turbines are designed to allow typically up to 5% of the compressor airflow for steam injection to the combustor and compressor discharge. The amount of steam injection is a function of gas turbine and gas turbine combustion system. Steam must contain at least 50°F (28°C) superheat and be at pressures comparable to fuel gas pressures. When either steam or water is used for power augmentation, the control system is normally designed to allow only the amount needed for NO_x abatement until the machine reaches base (full) load. At that point, additional steam or water can be admitted through the governor control.

Supplementary Fired HRSG Because gas turbines generally consume a small fraction of the available oxygen within the gas turbine air flow, the oxygen content of the gas turbine exhaust generally permits supplementary fuel firing ahead of (or within) the HRSG to increase steam production rates relative to an unfired unit. A supplementary fired unit is defined as an HRSG fired to an average temperature not exceeding about 1800°F (982°C). Because the turbine exhaust gas is essentially preheated combustion air, the supplementary fired HRSG fuel consumption is less than that required for a power boiler, providing the same incremental increase in steam generation. Incremental plant heat rate for supplementary firing is typically in the range of a simple-cycle gas turbine. An unfired HRSG with higher steam conditions is often designed with multiple pressure levels to recover as much energy as possible from the gas turbine exhaust. This adds cost to the unfired HRSG, but the economics are often enhanced for the cycle. In the case of the supplementary fired HRSG, if the HRSG is to be fired during most of its operating hours to the 1400-to-1800°F (760–982°C) range, then a suitably low stack temperature can usually be achieved with a single-pressure-level unit. This is the result of increased economizer duty as compared to the unfired HRSG. A supplementary fired HRSG has a design quite similar to that of an un-fired HRSG. However, the firing capability provides the ability to control the HRSG steam production within the capability of the burner system and independent of the normal gas turbine operating mode. Supplementary fired HRSGs are applicable to new units or combined-cycle add-ons. Retrofit installations on existing HRSGs are not practical due to the need for duct burner space and significant material changes. There is a small performance penalty when operating unfired compared to operating a unit designed without supplementary firing, and the magnitude of this performance penalty is directly proportional to the amount of supplementary firing built into the combined-cycle plant. The performance penalty is due to two factors: unfired operation results in lower stream flows and pressures and, thus, lower steam turbine efficiency; also, the pumps, auxiliary equipment and generator are sized for higher loads. Operating unfired results in comparatively higher parasitic loads compared to a unit designed solely for unfired operation. Peak Firing Users of some gas turbine models have the ability to increase their firing temperature above the base rating. This is known as peak firing, where both simple-cycle and combined-cycle output will increase. The penalty for this type of operation is shorter inspection cycles and increased maintenance. Despite this, running at elevated peak firing temperatures for short periods may be a cost-effective way to add kilowatts without the need for additional peripheral equipment.

Efficiency Enhancement Fuel Heating

If low-grade heat energy is available, this can be used to increase the temperature of gaseous fuels, which increases cycle efficiency by reducing the amount of fuel energy used to raise the fuel temperature to the combustion temperature. There is a very small (almost negligible) reduction in gas turbine output compared to the no-fuel heating case, primarily due to the lower gas turbine mass flow because of the reduction in fuel consumption. The reduction in combined-cycle output is typically greater than simple-cycle output primarily because the energy that would otherwise be used to make steam is often used to heat the fuel. Actual combined-cycle output and efficiency changes are dependent on fuel temperature rise and cycle design. Provided the fuel constituents are acceptable, fuel temperatures can potentially be increased up to approximately 700°F (370°C) before carbon deposits begin to form on heat transfer surfaces and the remainder of the fuel delivery system. For combined-cycle applications, fuel temperatures on the order of 300 to 450°F (150–230°C) are generally economically optimal. A combined-cycle plant has plenty of low-grade heat energy available. Typical F-class three-pressure reheat systems use water from the intermediate pressure economizer to heat the fuel to approximately 365°F (185°C). Under these conditions, efficiency gains of approximately 0.3 points can be expected for units with no stack temperature limitations. It is important to ensure that the fuel does not enter the steam system because maximum steam system temperatures are typically above the auto-ignition temperature for gas fuels. This can be accomplished in several ways. For a system utilizing a direct water-to-fuel heat exchanger, the water pressure is maintained above the fuel pressure so that any leakage takes place in the fuel system. Additional system design and operation requirements ensure that the fuel does not enter the steam system during periods when the water system is not a pressurized system. Other systems use an intermediate heat transfer fluid so that any fuel heat exchanger leakage cannot directly enter the steam system. For update opportunities, it must be considered that additional water flow may be required. Calculations must be performed to ensure the existing pump capabilities are not exceeded and that pressures are sufficient to deliver water to the HRSG drums under worst-case conditions. Other components that may see increased water flows (such as HRSG economizers) must be evaluated to ensure the design is acceptable.

Effect of Operating Parameters on the Performance of Combined Cycle Power Plant

The major operating parameters which influence the combined cycle performance are;

1. Turbine inlet temperature
2. Compressor pressure ratio
3. Pinch point
4. Ambient Temperature
5. Pressure levels

Effect of turbine inlet temperature

The power output of a gas turbine is a function of turbine inlet temperature. The turbine inlet temperature (TIT) plays an important role in the performance of combined cycle. The maximum value of TIT is fixed due to the metallurgical problem of turbine blade cooling. The results show that increases in the gas turbine inlet temperature decrease in the combustion chamber exergy destruction. The reason is due to the fact that this increase leads to the decrease of the entropy generation. The parameter that affects cycle performance most is the TIT (turbine inlet temperature). TIT should be kept on the higher side because at lower values, the exergy destruction is higher.

Effect of compressor pressure ratio

The temperature of air leaving the compressor and entering in combustion chamber depends upon the compressor pressure ratio. The increase in the compressor pressure ratio decreases the cost of exergy destruction. The reason is that by increasing the compressor ratio, the outlet temperature increases. So, the temperature difference decreases. Since the cost of exergy destruction is a direct function of exergy destruction. It leads to decrease in the cost of exergy destruction. As the compression ratio increases, the air exiting the compressors is hotter, therefore less fuel is required (lowering the air-fuel ratio) to reach the desired turbine inlet temperature, for fixed gas flow to the gas turbine. The work required in the compressor and the power output of the gas turbine steadily increases with compression ratio, then cause decreases in the exhaust gases temperature. This lower gas temperature causes less steam to be produced in the HRSG, therefore lowering the outputs of the steam cycle. It is noticed that the total power output increases with compression ratio. However, the variation of total power output is minor at lower compression ratio while it is significant at higher compression ratio for all gas turbine configurations.

Effect of the pinch point

The pinch point is the minimum difference between the gas temperature leaving the evaporator section of the HRSG and the saturation temperature corresponding to the steam pressure in the evaporator section. The effect of pinch point is that the increase of the pinch temperature of the heat recovery boiler, the cycle's exergy efficiency reduces, the reason is that the water section uses less of the energy of the output gases from the heat recovery boiler. Moreover, by increasing the temperature of the superheater, both the exergy loss in the heat recovery boiler and the cost of exergy loss are reduced. On the other hand, by raising the pinch temperature, both the exergy loss and the cost of exergy loss are increased.

CHAPTER-2: Introduction to the Project

2.1 - Gas Turbine

A gas turbine engine is a sort of internal combustion engine. Basically, the engine can be seen as an energy transformation gadget that transforms energy stored away in the fuel to helpful mechanical energy as rotational power. The expression "gas" alludes to the surrounding air that is taken into the engine and utilized as the working medium in the energy change process.

This air is first drawn into the engine where it is packed, blended with fuel and lighted. The subsequent hot gas grows at high speed through a progression of airfoil-formed sharp edges exchanging energy made from ignition to turn an output shaft. The remaining thermal energy in the hot exhaust gas can be bridled for an assortment of mechanical procedures.

Fundamental Components:

<u>Compressor</u>

The compressor takes in air and after that compacts and pressurizes the air atoms through a progression of pivoting and stationary compressor edges.

<u>Combustor</u>

In the combustor, fuel is added to the pressurized air particles and touched off. The warmed particles extend and move at high speed into the turbine segment.

<u>Turbine</u>

The turbine changes over the energy from the high-speed gas into valuable rotational power, however, extension of the warmed compacted gas over a progression of turbine rotor edges.

<u>Output Shaft and Gearbox</u>

Rotational power from the turbine segment is conveyed to driven hardware through the output shaft by means of a speed lessening gearbox.

Exhaust

The engine's exhaust area coordinates the spent gas out of the turbine segment and into the air.

Turbines types:

There is a wide range of sorts of turbines:

- You have most likely known about a **Steam turbine**. Most power plants utilize coal, flammable gas, oil or an atomic reactor to make steam. The steam goes through an immense and deliberately outlined multi-arrange turbine to turn a yield shaft that drives the plant's generator.
- Hydroelectric dams utilize **Water turbines** similarly to produce control. The turbines utilized as a part of a hydroelectric plant appear to be totally unique from a steam turbine since water is so significantly denser (and slower moving) than steam, however, it is a similar rule.
- **Wind turbines**, otherwise called windmills, utilize the wind as their thought process drive. A wind turbine looks not at all like a steam turbine or a water turbine since wind is moderate moving and light, however, once more, the rule is the same.

A gas turbine is an expansion of a similar idea. In a gas turbine, a pressurized gas turns the turbine. In every cutting-edge gas turbine motors, the motor delivers its own pressurized gas, and it does this by consuming something like propane, flammable gas, lamp oil or fly fuel. The warmth that originates from consuming the fuel extends air, and the fast surge of this hot air turns the turbine.

Working principle:

Gas turbine motors get their power from consuming fuel in an ignition chamber and utilizing the quick streaming burning gases to drive a turbine similarly as the high weight steam drives a steam turbine. A straightforward gas turbine is included in three fundamental areas a compressor, a combustor, and a power turbine. The gas-turbine works on the standard of the Brayton cycle, where the packed air is blended with fuel, and copied under steady weight conditions. The subsequent hot gas is permitted to extend through a turbine to perform work.

Gas Turbine Performance:

Certain climatic conditions critically affect any given gas turbine's accessible power:

a) Ambient temperature: As this ascents, a gas turbine may swallow a similar volume of air, yet that air will weigh less with expanding climatic temperature. Less air mass means less fuel mass is required to be touched off with that air and considerable lower power created.

b) Altitude: Increasing elevation implies bring down thickness air, so that is, in turn, diminishes power created by the turbine.

c) Humidity: Water vapor is less thick than air, so more water vapor in a given volume implies less weight of that air than if it had less water vapor. The impact is the same as with the two above elements.

The subject of performance optimization is an immense one which would incorporate a few subtopics. Channel cooling and water/steam infusion for power enlargement can be techniques which are utilized to supplement power "lost" by elements, for example, high ambient temperatures, and high elevation.

Contingent upon how one characterized performance optimization, the term could incorporate cycle modifications and emotionally supportive networks that are outside to the gas turbine center. A few cases are:

• Cycle modifications (which may likewise incorporate, however, are not restricted to, Delta cooling frameworks that are examined under "Outline Improvement")

• Engine condition observing frameworks

• Life cycle counters/assessment

2.2 – Recovery Steam Generator (HRSG) Heat

The heat recovery steam generator (HRSG) gives the thermodynamic connection between the gas turbines and steam turbines in a combined-cycle power plant. Each HRSG arrangement is exclusively designed to meet your coveted working adaptability and execution necessities.

A heat recovery steam generator (HRSG) is an energy recovery heat exchanger that recuperates heat from a hot gas stream. It produces steam that can be utilized as a part of a procedure (cogeneration) or used to drive a steam turbine (combined cycle).

HRSGs comprise four major components: **the economizer, evaporator, superheater** & **water preheater**. The distinctive parts are assembled to meet the working prerequisites of the unit. See the connected delineation of a Modular HRSG General Arrangement.

Modular HRSGs can be sorted by various courses, for example, heading of fumes gases stream or number of weight levels. In view of the stream of fumes gases, HRSGs are arranged into vertical and level composes. In level kind HRSGs, fumes gas streams on a level plane over vertical tubes while in vertical sort HRSGs, fumes gas stream vertically finished flat tubes. In light of weight levels, HRSGs can be ordered into single weight and multi-weight. Single weight HRSGs has just a single steam drum, and steam is created at single weight level though multi-weight HRSGs utilizes two (twofold weight) or three (triple weight) steam drums. All things considered, triple weight HRSGs comprises three areas: a LP (low weight) segment, a reheat/IP (moderate weight) segment, and a HP (high weight) segment. Each area has a steam drum and an evaporator segment where water is changed over to steam. This steam at that point goes through superheaters to raise the temperature past the immersion point.

The modular section in the outline classification is relied upon to hold a high piece of the overall industry before the finish of the evaluation time frame. In 2017, this portion mirrored a high piece of the overall industry of over half with a high market valuation of around US $449 million and is probably going to proceed with its ruling streak all through the time of evaluation to achieve a huge valuation before the year's over of appraisal (2027). It is anticipated to develop at a consistent CAGR amid the said period.

Package section in this classification is ready to develop at a generally high rate and is the second quickest developing fragment with a normal esteem CAGR of 8.6%. In any case, this portion has a low piece of the pie and valuation. The C-area section is the second biggest and is foreseen to pick up a 1.9x push in its incentive amid the 2017-2027 course of events. In 2017 it mirrored an estimation of above US$ 195 Mn securing its second place, after the modular portion.

2.3 – Co-generation Cycle

Cogeneration (cogen) through combined heat and power (CHP) is the simultaneous production of electricity with the recovery and utilization of heat. Cogeneration is a highly efficient form of energy conversion, and it can achieve primary energy savings of approximately 40% by compared to the separate purchase of electricity from the national electricity grid and a gas boiler for onsite heating.

Combined heat and power plants are typically embedded close to the end user and therefore help reduce transportation and distribution losses, improving the overall performance of the electricity transmission and distribution network for power users where security of supply is an important factor for their selection of power production equipment and gas is abundant, gas-based cogeneration systems are ideally suited as captive power plants (i.e., power plants located at site of use).

Benefits of Gas Engine CHP

The high efficiency of a CHP plant compared with conventional bought in electricity and site-produced heat provides a number of benefits including,

- On-site production of power
- Reduced energy costs
- Reduction in emissions compared to conventional electrical generators and onsite boilers Heat

Sources from a Gas Engine

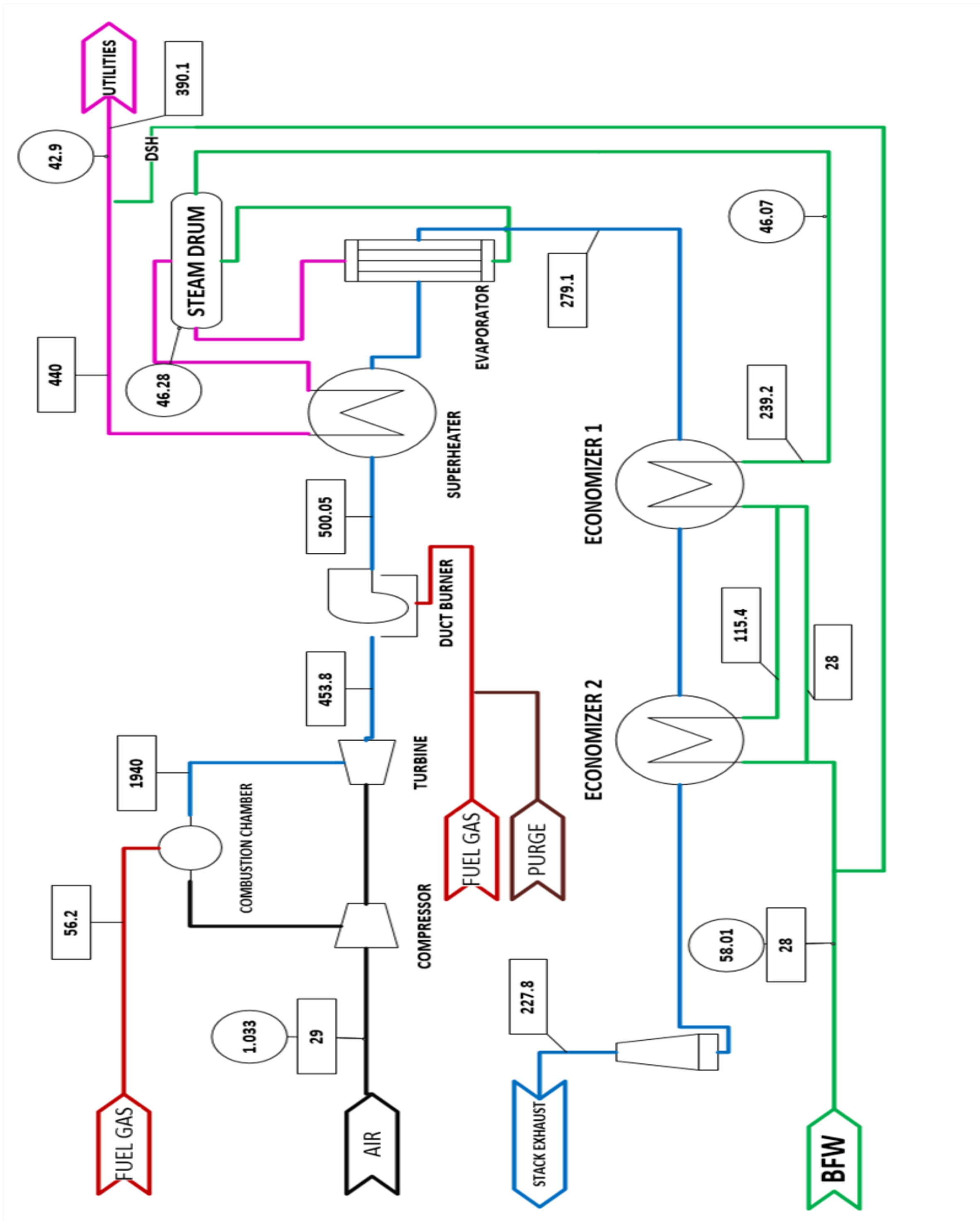
The heat from the generator is available in from 5 key areas:

1. Engine jacket cooling water
2. Engine lubrication oil cooling
3. First stage air intake intercooler
4. Engine exhaust gases
5. Engine generator radiated heat, second stage intercooler 1, 2 and 3 are recoverable in the form of hot water, typically on a 70/90˚C flow return basis and can be interfaced with the site at a plate heat exchanger.

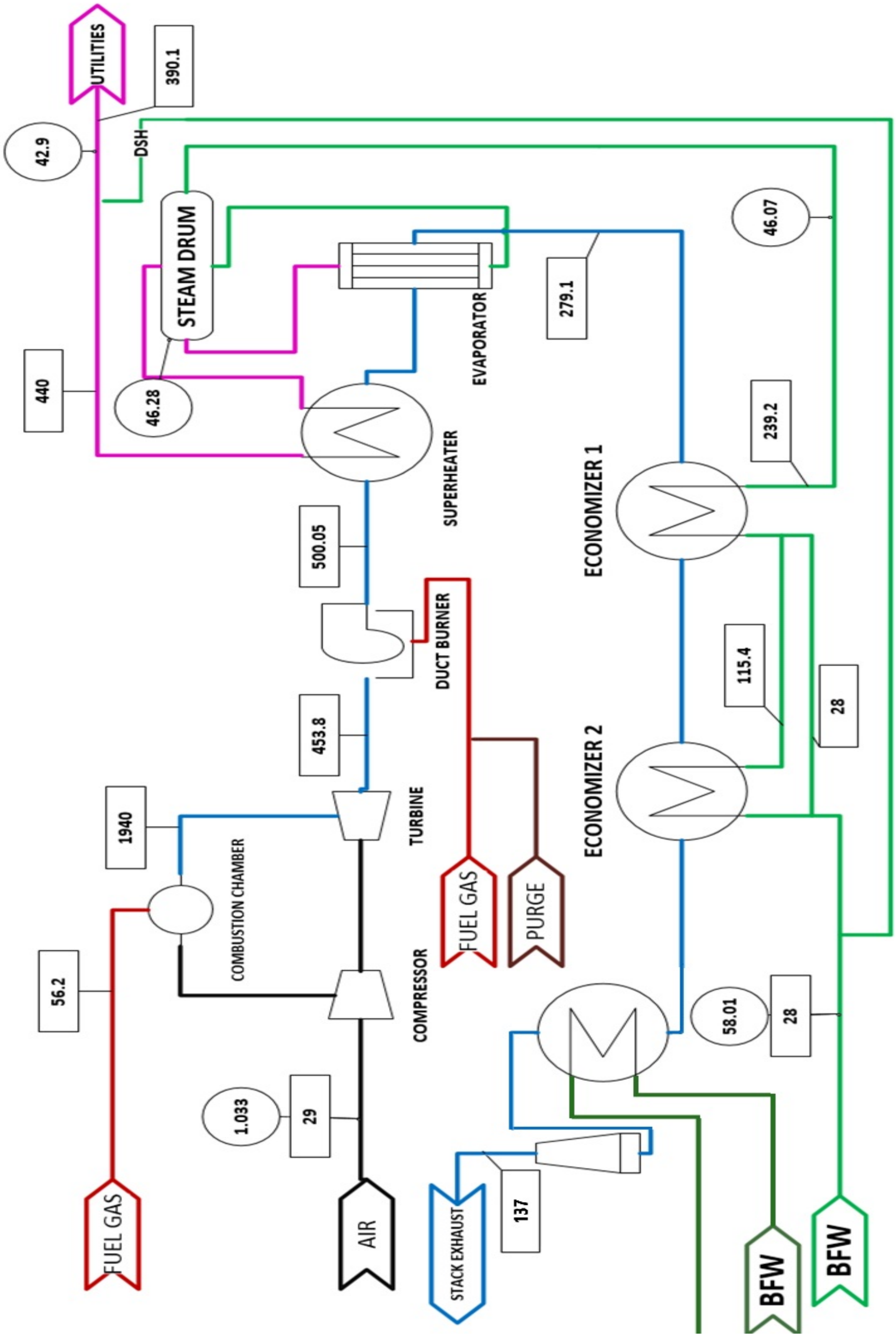
The engine exhaust gases typically leave the engine at between 400 and 500˚C. This can be used directly for drying, in a waste heat boiler to generate steam, or via an exhaust gas heat exchanger combining with the heat from the cooling circuits. The heat from the second stage intercooler is also available for recovery as a lower grade heat. Alternatively, new technologies are available for the conversion of heat to further electricity, such as the Organic Rankine Cycle Engine. CHP applications. A variety of different fuels can be used to facilitate cogeneration. In gas engine applications CHP equipment is typically applied to natural gas (commercial, residential and industrial applications), biogas and coal gas applications. CHP System Efficiency Gas engine combined heat and power systems are measured based upon the efficiency of conversion of the fuel gas to useful outputs. The diagram below illustrates this concept. Firstly the energy in the fuel gas input is converted into mechanical energy via the combustion of the gas in the engine's cylinders and their resulting action in the turning of the engine's crankshaft. This mechanical energy is in turn used to turn the engine's alternator in order to produce electricity.

2.4 – Process Flow Diagram

2.4.1–Pre-Enhancement:

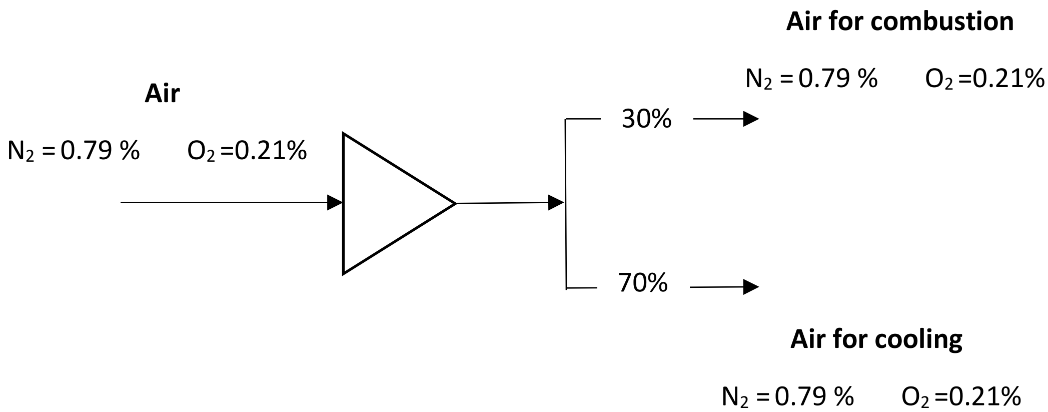


2.4.2–Post-Enhancement:



CHAPTER-3: Material Balance

3.1–Material Balance on compressor:



kgmol of inlet air = 207796.03

kgmol of air for combustion + kgmol of air for cooling = total outlet air from compressor

62338.8 + 145457.22 = 207796.03

Conversion of $^{NMCH}/_{kg/cm2}$ into $kg\ mole/hr$:

Fuel gas:

Components	% Composition
C_2H_6	0.09
CH_4	76.97
C_6H_{14}	0.02
CO_2	0.39
N_2	22.53

Average molecular weight = 18.84

Ideal gas constant = 84.784 × 10^{-6} $m^3 kgf/cm^2 Kmol$

$$n = \frac{PV}{RT}$$

$$= \frac{1.033\ ^{kgf}/_{cm^2} \times 5843.8\ ^{m^3}/_{hr}}{84.784 \times 10{-}6\ m^3 kgf/cm^2 Kmol \times 273\ K}$$

$= 2.61 \times 10^5\ g\ mole/hr$
$= \mathbf{260\ kg\ mole/hr}$
$= 260 \times 18.84 = 4912\ kg/hr$

Purge gas:

Components	% Composition
H_2	12.82
CH_4	22.6
Ar	7.9
N_2	56.6

Average molecular weight = 22.87
Ideal gas constant = $84.784 \times 10^{-6}\ m^3 kgf/cm^2 Kmol$

$$n = \frac{PV}{RT}$$
$$= \frac{1.033\ {}^{kgf}/_{cm^2} \times 1668.6\ {}^{m^3}/_{hr}}{84.784 \times 10{-}6\ m^3 kgf/cm^2 Kmol \times 273\ K}$$
$= 74469.08\ g\ mole/hr$
$= \mathbf{74.47\ kg\ mole/hr}$
$= 74.47 \times 22.87 = 1703.10\ kg/hr$

3.2–Material Balance on combustion Chamber:

Objective: 16% excess oxygen at outlet

Fuel gas enters the combustion chamber at 29°C and 1.033 kgf/cm^2.
The flowrate of the fuel gas is 5843.8 ${}^{NMCH}/_{kg/cm^2}$
Assuming the combustion is for all hydrocarbons is complete except methane with combustion 88%.
The reactions occurring in the turbine are

1: $CH_4 + 2O_2 \longrightarrow CO_2 + 2H_2O$
2: $C_2H_6 + \frac{7}{2}O_2 \longrightarrow 2CO_2 + 3H_2O$
3: $C_6H_{14} + \frac{19}{2}O_2 \longrightarrow 6CO_2 + 7H_2O$
4: $CH_4 + O_2 \longrightarrow 2CO + 4H_2O$

Fuel inlet composition of are following:

Components	% Composition
C_2H_6	0.09
CH_4	76.97
C_6H_{14}	0.02
CO_2	0.39
N_2	22.53

30% of air from compressor enters the combustion chamber the composition is as follow:

Components	%Composition
N_2	79
O_2	21

Basis:

260 $kg\ mole/hr$ of feed gas

Reaction 1:

$$\mathbf{O_2\ needed} = \frac{76.97}{100} \times 260 = 200.12$$

$$= 200.12 \times 2 = 400.244 \text{ kg moles}$$

88 % combustion efficiency = 400.244 × 0.88 = 352.21 kg moles

Reaction 2:

$$\mathbf{O_2\ needed} = \frac{0.09}{100} \times 260 = 0.234$$

$$= \frac{7}{2} \times 0.234 = 0.819 \text{ kg moles}$$

Reaction 3:

$$\mathbf{O_2\ needed} = \frac{0.02}{100} \times 260 = 0.052$$

$$= \frac{19}{2} \times 0.052 = 0.494 \text{ kg moles}$$

Reaction 4:

$$\mathbf{O_2\ needed} = \frac{after\ 88\%\ combustion + before\ 88\%\ combustion}{reactant\ oxygen}$$

$$= \frac{-352.21+400.244}{2} = 24.017$$

$$= \frac{3}{2} \times 24.017 = 36.0255 \text{ kg moles}$$

Total O_2 needed for combustion = 389.5485 kg moles

1) **<u>CO_2 Produced:</u>**

<u>Reaction 1:</u>

CO_2 Produced $= 200.122 \times 0.88$
$= 176.11$ kg moles

<u>Reaction 2:</u>

CO_2 Produced $= 0.234 \times 2$
$= 0.468$ kg moles

<u>Reaction 3:</u>

CO_2 Produced $= 0.052 \times 6$
$= 0.312$ kg moles

Total CO_2 produced = 176.6092 kg moles

2) **<u>H_2O Produced:</u>**

<u>Reaction 1:</u>

H_2O Produced $= 200.122 \times 0.88 \times 2$
$= 352.215$ kg moles

<u>Reaction 2:</u>

H_2O Produced $= 0.234 \times 3$
$= 0.702$ kg moles

<u>Reaction 3:</u>

H_2O Produced $= 0.052 \times 7$
$= 0.364$ kg moles

<u>Reaction 4:</u>

H_2O Produced = 24.017×2
= 48.034 kg moles

Total H_2O Produced = 401.315 kg moles

3) <u>Air Needed:</u>

Total O_2 = 389.5485 kg moles

O_2 needed = $\frac{\text{O2 in } feed + therotical \text{ O2 needed}}{therotical \text{ O2 needed}}$

$$\frac{16}{100} = \frac{\text{O2 in } feed + 389.5485}{389.5485}$$

O2 in feed = 451.878 kg moles

Air needed = $\frac{451.878}{0.21}$
= 2151.8 kg moles

N_2 in air = 2151.8×0.79
= 166.922 kg moles

Total N_2 = 1758.5 kg moles

4) <u>CO Needed:</u>

CO Produced = 24.017 kg moles

5) <u>Overall Material Balance:</u>

<u>*Inlet:*</u>

<u>Fuel Gas:</u>

CH_4 = 200.122×16 = 3201.952 kg
N_2 = 58.578×28 = 1640.2 kg
CO_2 = 1.014×44 = 44.616 kg

C_2H_6 = 0.234 × 30 = 7.02 kg
C_6H_{14} = 0.052 × 86 = 4.472 kg

Total Fuel Gas = 4898.26 kg

<u>Air:</u>
O_2 = 451.878 × 32 = 14460.16 kg
N_2 = 1758.5 × 28 = 49238 kg

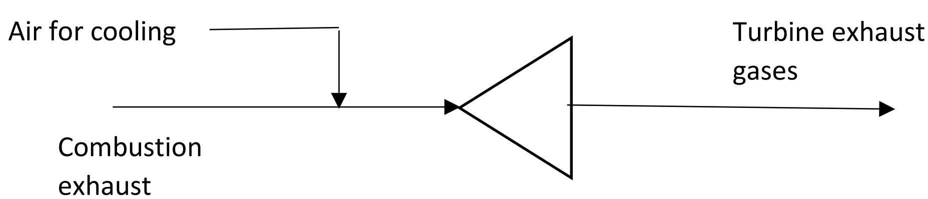
Total Air = 63698.16 kgs
Total Inlet = 68596.42 kgs

<u>*Outlet*:</u>

O_2 = 451.878 – 389.5485 = 62.33 × 32 = 1994.5 kg
N_2 = 1758.5 × 28 = 49238 kg
CO_2 = 176.6092 × 44 = 7770.8 kg
CO = 0.234 × 30 = 7.02 kg
H_2O = 401.315 × 18 = 7223.67 kg

Total Inlet = 66899.446 kgs

3.3–Material Balance on combustion Chamber:



Air for cooling + combustion exhaust = turbine exhaust gases
Total air = 62338.8 / 0.3 = 207796.03 kg
Air for cooling = total air × 0.7
= 145457.22 kg
Turbine exhaust gases = 145457.22 + 66956.99
= 212414.12 kg

3.4–Material Balance on Heat Recovery Steam Generator:

Purge gas flowrate = 96.88 $kg\ mole/hr$
Fuel gas flowrate = 0.674 $kg\ mole/hr$
Turbine exhaust gas flowrate = 7475 $kg\ mole/hr$

For turbine exhaust gases:

Components	Composition	flowrate
O_2	0.1502	1122.7450
CO_2	0.0238	117.9805
CO	0.0032	23.8528
H_2O	0.0537	401.4075
N_2	0.7691	5749.0225

For turbine exhaust gases:

Components	Composition	flowrate
CH_4	0.7697	0.5187
C_2H_6	0.0009	0.0006
CO_2	0.0039	0.0026
C_6H_{14}	0.0002	0.00013
N_2	0.2253	0.1518

For purge gas:

Components	Composition	flowrate
H_2	0.1282	12.42
Ar	0.079	7.653
CH_4	0.566	54.834
N_2	0.226	21.894

Total CH_4 = 21.894 + 0.5187 = 22.41227

1: $CH_4 + 2O_2 \longrightarrow CO_2 + 2H_2O$
2: $C_2H_6 + \frac{7}{2}O_2 \longrightarrow 2CO_2 + 3H_2O$
3: $C_6H_{14} + \frac{19}{2}O_2 \longrightarrow 6CO_2 + 7H_2O$
4: $H_2 + O_2 \longrightarrow 2H_2O$

Assuming complete combustion

<u>O_2 consume:</u>

<u>Reaction 1:</u>

O_2 consume = 2×22.4127
= 44.8254

Reaction 2:

O_2 consume = $^7/_2 \times 0.0006$

= 0.0021

Reaction 3:

O_2 consume = $^{19}/_2 \times 0.00013$

= 0.001235

Reaction 4:

O_2 consume = $^1/_2 \times 12.42$

= 6.21

Total O_2 consume = 51.038 kg mole

CO_2 Produced:

Reaction 1:

CO_2 consume = 22.4127

= 22.4127

Reaction 2:

CO_2 consume = 2 × 0.0006

= 0.0012

Reaction 3:

CO_2 consume = 6 × 0.00013

= 0.00078

Total CO_2 consume = 22.4147 kg mole

H_2O Produced:

Reaction 1:

H_2O consume = 2 × 22.4127

= 44.8254

Reaction 2:

H_2O consume = 3 × 0.0006

= 0.0018

Reaction 3:

H_2O consume = 7 × 0.00013

= 0.00091

Reaction 4:

H_2O consume = 12.42

Total H_2O consume = 57.24811 kgmole

Exhaust gases from HRSG

CO_2 = 117.98050 + 22.4147 + 0.0026 = 140.3978

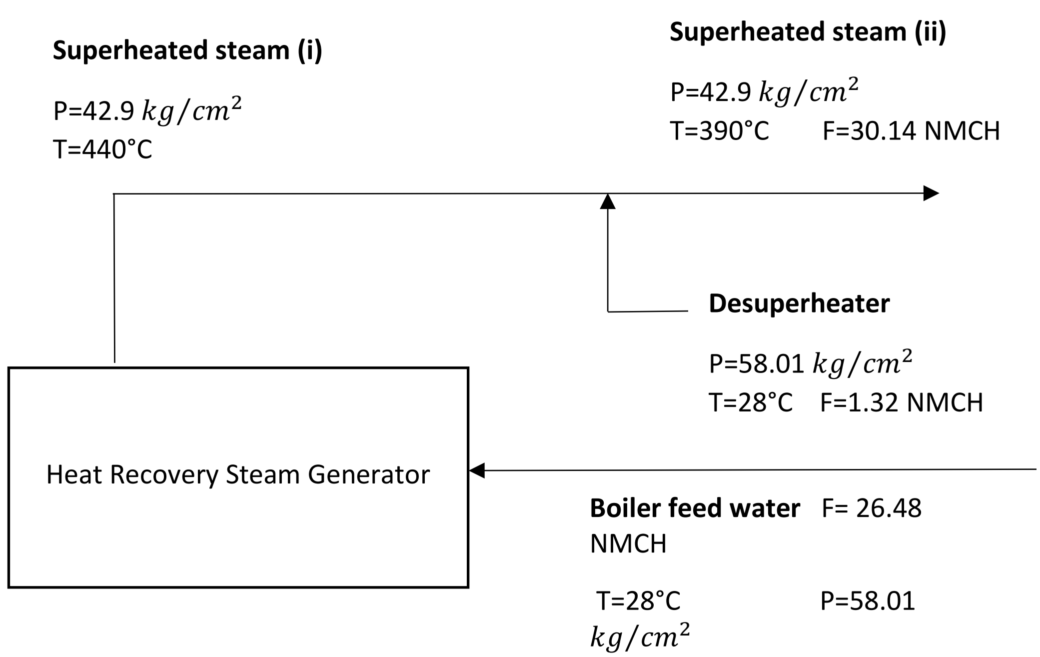
CO = 23.8528
H_2O = 401.4075 + 57.24811 = 458.65561
O_2 = 1122.745 – 51.038 = 1071.707
N_2 = 5749.0225 + 0.1518 + 54.834 = 5804.0083
Ar = 7.653

Total outlet of HRSG = 7506.27451 kgmole

<u>Compositions:</u>

CO_2 = 0.0187
CO = 0.0032
H_2O = 0.06110
O_2 = 0.1428
N_2 = 0.7732
Ar = 0.0010

Water Balance:



<u>Boiler feed water:</u>

F = 26.48 NMCH
ρ = 1007 kg/m^3
$\rho = \frac{m}{V}$
m = $\rho \times$ V
= 1007 $kg/m^3 \times 26.48\ m^3/hr$
= 26665.36 kg

n = 1479.76 kgmol

Desuperheater:

F = 1.32 NMCH

ρ = 1007 kg/m^3

$\rho = \frac{m}{V}$

m = $\rho \times$ V

= 1007 $kg/m^3 \times$ 1.32 m^3/hr

= 1329.24 kg

n = 73.76 kgmol

moles of water = moles of steam (superheated steam (i))

1479.76 = 1479.76

Total moles at outlet superheated steam (ii)

superheated steam (i) + Desuperheater = superheated steam (ii)

1479.76 + 72.76 = 1553.52 (outlet steam)

Total volumetric flowrate at outlet of superheated steam (ii)

Compressibility factor = Z = 0.93

ρ = 14.76 kg/m^3

n = 1553.52 kgmole

m = 27994.6 kg

$\rho = \frac{m}{V}$

V= $\frac{m}{\rho}$

V= $\frac{27994.6}{14.76}$

= 1896.65 m^3/hr

CHAPTER-4: Energy Balance

4.1–Energy Balance on Compressor:

The overall energy balance equation

Q_{in} + $\widehat{H_1}$ + W + gh = Q_{out} + $\widehat{H_2}$

Q_{in} = 0 gh = 0

$\dot{m}\ \widehat{H_1}$ + W_{in} = Q_{losses} + $\dot{m}\ \widehat{H_2}$

H_1 = 302.63 kJ/kg H_2 = 575.9 kJ/kg

T_1 = 29°C T_2 = 300°C

η = 0.68

$\dot{m}$ = 2.078 × 10^5 kg/hr = 57.72 kg/s

<u>Given</u>

60 to 65% power of whole equipment is used by the compressor. The turbine is 9MW so 62% is used by the compressor.

W_{in} = 9MW × 0.62

= 5.58 × 10^6 Watts

Q_{losses} = -$\dot{m}\ \widehat{H_2}$ + $\dot{m}\ \widehat{H_1}$ + W_{in}

= $\dot{m}$ ($\widehat{H_1}$ - $\widehat{H_2}$) + W_{in}

= 57.72 (302.63 – 575.9) + 5580

= -10193.14 kW

4.2–Energy Balance on Combustion Chamber:

<u>Air:</u>

$C_{P\ air}$ = 30.9 kJ/kgC

<u>Fuel:</u>

$C_{P\ fuel}$ = 36.2 kJ/kgC

T_{inlet} = 300°C

$\dot{m}_{in}$= $\dot{m}_{out}$= 2161 kg mole

T_{fuel} = 56.2°C

$\dot{m}_{in}$= $\dot{m}_{out}$= 260 kg mole

Input:

$Q = mC_{P\,air}\,\Delta T + \; mC_{p\,fuel}\,\Delta T$

= (2161) (36.2) (300 – 25) + (260) (36.2) (56.2 – 25)

Q_{in}= 18.656 MJ

Heat of Reaction = -585.790 $kJ/kgmol$

n = 260 moles

$Q_{produced}$ = -585.790 $kJ/kgmol$ × 260 moles

= -152.3 MJ

Output:

$C_{P\,out}$ = 44.10 kJ/kgC

T_{out} = 1841°C

$\dot{m}_{in}$= $\dot{m}_{out}$= 2161 kg mole

$Q = mC_{P\,out}\,\Delta T$

= (2161) (44.10) (1841 – 25)

Q_{out}= 173.069 MJ

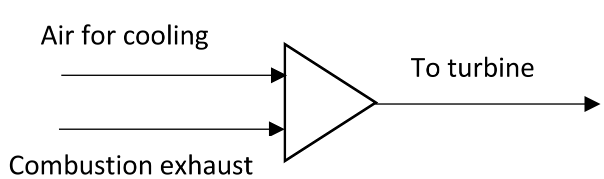
Overall balance:

Q_{in} - $Q_{produced}$ = Q_{out}

18.656 MJ – (-152.3 MJ) = 173.069 MJ

170.96 MJ = 173.069 MJ

4.3–Energy Balance on mixer:



$C_{P\,cooling\,air}$= 30.9$\frac{kJ}{kgC}$ $C_{P\,combustion\,exhaust}$= 44.1$\frac{kJ}{kgC}$ $C_{P\,to\,turbine}$= 34.79$\frac{kJ}{kgC}$

$T_{cooling\ air}$ = 300°C $T_{combustion\ exhaust}$ = 1940°C $T_{to\ turbine}$ = 909.1°C

$Q = C_P\, \Delta T = m\widehat{H_{in}}$

$\widehat{H_{cooling\ air}}$ = 30.9 (300 – 25) = 8.49 × 10³ kJ/mol

$Q_{cooling\ air}$ = 5042.40 mol × 8.49 × 10³ kJ/mol

$$\boldsymbol{Q_{cooling\ air} = 4.281 \times 10^7\ kJ}$$

$\widehat{H_{combustion\ exhaust}}$ = 44.1 (1940 – 25) = 8.44 × 10³ kJ/mol

$Q_{combustion\ exhaust}$ = 24289.09 mol × 8.44 × 10³ kJ/mol

$$\boldsymbol{Q_{combustion\ exhaust} = 2.05 \times 10^8\ kJ}$$

$\widehat{H_{to\ turbine}}$ = 34.79 (909.1 – 25) = 3.076 × 10⁴ kJ/mol

$Q_{to\ turbine}$ = 7473.99 mol × 3.076 × 10⁴ kJ/mol

$$\boldsymbol{Q_{to\ turbine} = 2.299 \times 10^8\ kJ}$$

4.4–Energy Balance on turbine:

Turbine inlet gases:

Components	Composition	flowrate
O_2	0.1502	1122.7450
CO_2	0.0238	117.9805
CO	0.0032	23.8528
H_2O	0.0537	401.4075
N_2	0.7691	5749.0225

Mass flowrate = $\dot{m}$ = 2.127 × 10⁵ kg/hr

= 59.08 kg/s

T_{inlet} = 909.1°C T_{outlet} = 453.8°C

$C_{P\ in}$ = 1.2225 kJ/kgC $C_{P\ out}$ = 1.139 kJ/kgC

H = m$C_P\, \Delta T$

$\dot{m}\ \widehat{H_{in}}$ = 59.08 kg/s × 1.2225 kJ/kgC (909.1- 25)

= 63828.27 kJ/s = 6.383 × 10⁴ kW

$\dot{m}\ \widehat{H_{out}}$ = 59.08 kg/s × 1.139 kJ/kgC (453.8- 25)

$= 30960861.06\ kJ/s = 3.096 \times 10^4$ kW

Given $W_{out} = 9 \times 10^6$ Watts

Overall energy balance equation

$Q_{in} + \dot{m}\,\widehat{H_1} + W_{in} = Q_{out} + \dot{m}\,\widehat{H_2} + W_{out}$

$Q_{in} = 0$ $\qquad\qquad\qquad W_{in} = 0$

$Q_{out} = \dot{m}\,\widehat{H_1} - \dot{m}\,\widehat{H_2} - W_{out}$

$= 6.383 \times 10^4$ kW - 3.096×10^4 kW – 0.9×10^4 kW

Q_{out} = 2.387 Kw

4.5–Energy Balance on heat recovery steam generator:

<u>For gases:</u>

$C_{P\,in} = 32.97\ kJ/kgC$ $\qquad\qquad C_{P\,out} = 31.00\ kJ/kgC$

$T_{inlet} = 541.1$°C $\qquad\qquad T_{outlet} = 227$°C

$\dot{m}_{in} = \dot{m}_{out} = 7566$ kg mole

<u>Inlet:</u>

$Q = C_P\,\Delta T = \quad m\widehat{H_{in}}$

$\widehat{H_{in}} = 32.97\ (541.1 - 25)$

$= 17015.82\ kJ/kg\,mol$

$Q_{in} = 7566 \times 17015.82 = 1.2874 \times 10^8$ kJ

<u>Outlet:</u>

$Q = C_P\,\Delta T = \quad m\widehat{H_{out}}$

$\widehat{H_{out}} = 31.00\ (227 - 25)$

$= 6262\ kJ/kg\,mol$

$Q_{out} = 7566 \times 6262 = 4.738 \times 10^7$ kJ

$\boldsymbol{\Delta Q = Q_{out} - Q_{in}}$

$= 80.697 \times 10^6$ kJ

This is the heat released by the hot flue gases and used by water to produce steam.

For Water:

$C_{P\ in}$ = 79.11 kJ/kgC $C_{P\ out}$ = 40.92 kJ/kgC

T_{inlet} = 28°C T_{outlet} = 440°C

$\dot{m}_{in} = \dot{m}_{out}$ = 1479.76 kg mole

Inlet:

The boiling point of water at 46.28 kg/cm^2 is 258.1°C

$Q = C_P\,\Delta T = m\widehat{H_{in}}$

$\widehat{H_{in}}$ = 79.11 (28 – 25)

= 304.8 $kJ/kg\,mol$

Q_{in} = 1479.76 × 304.8 = 4510.308 × 10^2 kJ

$H_{in\ \widehat{saturated}\ steam}$ = 101.6 (258.1 – 25)

= 23682.96 $kJ/kg\,mol$

$Q_{saturated\ steam}$ = 1479.76 × 23682.96 = 35.04 × 10^6 kJ

$\Delta Q_{saturated\ steam}$ = 34.59 × 10^6 kJ

Outlet:

$Q = C_P\,\Delta T = m\widehat{H_{out}}$

$\widehat{H_{out}}$ = 40.92 (440 – 258.1)

= 7443.35 $kJ/kg\,mol$

Q_{out} = 7566 × 7443.35 = 11.014 × 10^6 kJ

Heat of vapourization:

a = 52.053 T_c= 647.13°C n = 0.321

$\Delta H_V = a \times (1 - \frac{T}{T_c})^n$ = 23413.93 $kJ/kg\,mol$

Q_L = 3.484 × 10^7 kJ

Q_{water} = 3.484 × 10^7 + 11.014 × 10^6 + 34.59 × 10^6

= 80.44 × 10^6 kJ

CHAPTER-5: Efficiency Calculations

5.1–Efficiency calculations for compressor:

Isentropic efficiency:

Inlet:

P= 1.01325 bars

T_{inlet} = 29°C = 302 K

outlet:

P=5.78 bars

T_{outlet} = 300.3°C = 573.3 K

Calculated outlet Temperature formula:

$\acute{T}_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{R}{C_{P\,mean}}}$

Temperature is in kelvin

R= C_P - C_V

$\acute{T}_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{C_P - C_V}{C_P}}$

$\acute{T}_2 = T_1 \left(\frac{P_2}{P_1}\right)^{1-\frac{1}{r}}$

r = $\frac{C_P}{C_V}$

$\acute{T}_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{r-1}{r}}$

r_{in} = 1.398

r_{out} = 1.368

r_{mean} = 1.383

$\acute{T}_2 = 302 \left(\frac{5.78}{1.10325}\right)^{\frac{1.383-1}{1.383}}$

$\acute{T}_2$= 486.2 K = 213.18°C

This is the isentropic outlet temperature.

Properties of Air

Inlet:

$C_{P\,in}$ = 1.007 kJ/kgK

$\widehat{S_{in}}$ = 6.877 kJ/kgK

$\widehat{H_{in}}$ = 302.63 kJ/kgK

ρ = 1.169 kg/m^3

Outlet:

$C_{P\,out}$ = 1.11 kJ/kgK

$\widehat{S_{out}}$ = 7.46 kJ/kgK

$\widehat{H_{out}}$ = 874.18 kJ/kgK

ρ = 2.38 kg/m^3

$\widehat{H_2}$ =at actual outlet temperature = 575.9 kJ/kg

$\acute{H_2}$= calculated outlet temperature = 488.98 kJ/kg

H_1 = 302 kJ/kg

Isentropic η = $\frac{\acute{H_2}-H_1}{H_2-H_1} \times 100$

$= \frac{488.98-302}{575.9-302} \times 100$

η = 68.3%

<u>Polytropic efficiency</u>

$$\eta_{poly} = \frac{\frac{k-1}{k} \times ln\frac{P_{dis}}{P_{suc}}}{ln\frac{T_{dis}}{T_{suc}}}$$

Pressure is in psia.

P_{dis} = 82.21 psia P_{suc} = 14.65 psia

T_{dis} = 29°C = 543.87 R T_{suc} = 300.2°C = 1031.67 R

k = $\frac{C_P}{C_V}$ = 1.398

$$\eta_{poly} = \frac{\frac{1.398-1}{1.398} \times ln\frac{82.21}{14.65}}{ln\frac{1031.67}{543.87}} = 0.767 = 76.7\ \%$$

Polytropic η of compressor is greater than isentropic η.

Combustion η = 88%

Compressor η = 76.7%

Turbine η = 91.68 %

Overall η = 0.88 × 0.767 × 0.9168 = 61.88 %

5.2–Efficiency calculations for Turbine:

<u>Isentropic efficiency:</u>

$$\eta_{turbine} = \frac{H_{inlet}-H_{\acute{outlet}}}{H_{inlet}-H_{outlet}}$$

In terms of temperature and pressure:

$$\eta_{turbine} = \frac{T_{inlet}-T_{outlet}}{T_{inlet}\left((1-\frac{1}{P_{in}/P_{out}})^{\frac{r-1}{r}}\right)}$$

$r = \frac{C_P}{C_V}$

r_{in} = 1.314 r_{out} = 1.346 r_{mean} = 1.33

$T_{insentropic} = T_{in}(\frac{P_{out}}{P_{in}})^{\frac{k-1}{k}}$ = 440.9°C

Now

$\eta_{turbine} = \frac{Actual\ work}{isentropic\ work}$

$\dot{m} = 2.127 \times 10^5\ kg/hr = 59.08\ kg/s$

So,

$H_{in} = mC_P\,\Delta T$ = 59.08 × 1.222 (909.1- 25)

$\widehat{H_{in}}\dot{m}$ = 63828.27 kJ/s = 6.383×10^4 kW

$\widehat{H_{out}}\dot{m}$ = 59.08 × 1.139 (453.8 - 25)

$\widehat{H_{out}}\dot{m}$ = 3.096×10^4 kW (Actual)

$\widehat{H_{isen\ IN}}\dot{m}$ = 59.08 × 1.139 (440.9 - 25

= 27986.8 Kw

$\widehat{H_{isen\ OUT}}\dot{m}$ = 2.798×10^4 kW

$$\eta = \frac{\widehat{H_{in}}\dot{m} - \widehat{H_{out}}\dot{m}}{\widehat{H_{in}}\dot{m} - \widehat{H_{isen\ OUT}}\dot{m}} = \frac{6.383 \times 10^4 - 3.096 \times 10^4}{6.383 \times 10^4 - 2.798 \times 10^4}$$

η = 91.68%

<u>Thermal efficiency turbine:</u>

Flowrate = 5843.8 NMCH of gas

Conversion

1 scf = 1.61 Nm^3/hr

37.28 SCFH = 1 NMCH

So, gas flow = 217781.4 SCFH

A heating value for the given composition of gas = 680 $BTU/scfh$

Total heat input = 217781.4 SCFH × 680$BTU/SCFH$

$\Delta Q_{in} = 148.091 \times 10^6\ BTU/hr$ (by combustion)

power output of turbine:

W_{out} = 9MW

Conversion:

1MW = 341441.63 BTU/hr

9MW = 3.071×10^7 BTU/hr

$\eta_{thermal} = \frac{3.071 \times 10^7 \ BTU/hr}{148.091 \times 10^6 \ BTU/hr}$

= 0.2073

$\eta_{thermal}$ = 20.73%

5.3–Efficiency calculations for HRSG:

Economizer 1

$T_{in\ gas} = 269.5°C$ $\quad$ $T_{out\ gas} = 227°C$

$T_{in\ water} = 28°C$ $\quad$ $T_{out\ water} = 115.4°C$

C_C = 26.48 m^3/hr

= 26.48 / 3600 s

= 0.00735 $m^3/s \times 947.9\ kg/m^3$ = 6.97 kg/s

C_C = 6.97 × 4182 = 29158.28 W/C

q = 29158.28 (115.4 – 28) = 2.25×10^6 W

$C_h = \frac{q}{\Delta T} = \frac{2.25 \times 10^6}{269.5-227}$ = 59963.14 W/C

η NTU= $\frac{q}{C_{\min}(\overline{T}-\overline{t})}$

$\overline{T} = \frac{269.5+227}{2}$ = 248.25 $\quad$ $\overline{t} = \frac{115.4+28}{2}$ = 71.1

η NTU= $\frac{2.25 \times 10^6}{29158.28\ (248.25-71.7)}$ = 0.4953

$$NTU = \frac{tanh^{-1}(\eta\ NTU \times \frac{\sqrt{1+C_r^2}}{2}}{\frac{\sqrt{1+C_r^2}}{2}}$$

$C_r = \frac{29158.28}{59963.14}$ = 0.4862

$$\text{NTU} = \frac{tanh^{-1}(0.4953 \times \frac{\sqrt{1+0.4862}}{2}}{\frac{\sqrt{1+0.4862}}{2}}$$

NTU = 0.508

η NTU= 0.4953

$$\eta = \frac{0.4953}{NTU} = \frac{0.4953}{0.508}$$

η = 97.44%

Economizer 2

$T_{in\ gas} = 336.3\ °C$ $T_{out\ gas} = 269.5°C$

$T_{in\ water} = 112°C$ $T_{out\ water} = 239.2°C$

C_C = 26.48 m^3/hr

= 26.48 / 3600 s

= 0.00735 m^3/s × 947.9 kg/m^3 = 6.97 kg/s

C_C = 6.97 × 4182 = 29158.28 W/C

q = 29158.28 (239.2 – 112) = 3.708 × 10^6 W

$$C_h = \frac{q}{\Delta T} = \frac{3.708 \times 10^6}{336.3-269.5} = 55522.95\ W/C$$

$$\eta\ \text{NTU} = \frac{q}{C_{\min}(\overline{T}-\overline{t})}$$

$$\overline{T} = \frac{336.3+269.5}{2} = 302.9 \qquad \overline{t} = \frac{239.2+112}{2} = 112$$

$$\eta\ \text{NTU} = \frac{3.708 \times 10^6}{29158.28\ (302.9-175.6)} = 0.998$$

$$\text{NTU} = \frac{tanh^{-1}(\eta\ \text{NTU} \times \frac{\sqrt{1+C_r^2}}{2}}{\frac{\sqrt{1+C_r^2}}{2}}$$

$$C_r = \frac{29158.28}{55522.95} = 0.5251$$

$$\text{NTU} = \frac{tanh^{-1}(0.998 \times \frac{\sqrt{1+0.5251}}{2}}{\frac{\sqrt{1+0.5251}}{2}}$$

NTU = 1.129

η NTU= 0.998

$\eta = \frac{0.998}{NTU} = \frac{0.998}{1.129}$

η = 88%

<u>Evaporator:</u>

$T_{in\ gas} = 423.3°C$ $T_{out\ gas} = 336.3°C$

$T_{in\ water} = 239.4°C$ $T_{out\ water} = 270°C$

$C_C = 26.48\ m^3/hr$

$= 26.48 / 3600$ s

$= 0.00735\ m^3/s \times 947.9\ kg/m^3 = 6.97\ kg/s$

$C_C = 6.97 \times 4182 = 29158.28\ W/C$

$q = 29158.28\ (270 - 239.4) = 8.92 \times 10^5$ W

$C_h = \frac{q}{\Delta T} = \frac{8.92 \times 10^5}{423.3-336.3} = 33825\ W/C$

Heat of vaporization = 243800

Total heat = 3.33×10^6 W

$\eta\ NTU = \frac{q}{C_{\min}(\overline{T}-\overline{t})}$

$\overline{T} = \frac{423.3-336.3}{2} = 379.8$ $\overline{t} = \frac{270+239.2}{2} = 246.8$

$\eta\ NTU = \frac{3.33 \times 10^6}{29158.28\ (379.8-246.8)} = 0.8601$

$$NTU = \frac{tanh^{-1}(\eta\ NTU \times \frac{\sqrt{1+C_r^2}}{2}}{\frac{\sqrt{1+C_r^2}}{2}}$$

$C_r = \frac{29158.28}{33825.98} = 0.862$

$$NTU = \frac{tanh^{-1}(0.903 \times \frac{\sqrt{1+0.862}}{2}}{\frac{\sqrt{1+0.862}}{2}}$$

NTU = 1.04

η NTU= 0.8601

$\eta = \frac{0.4953}{NTU} = \frac{0.8601}{0.903}$

η = 82.8%

Superheater:

$T_{in\ gas} = 541.1°C$ $\quad$ $T_{out\ gas} = 423°C$

$T_{in\ water} = 254.4°C$ $\quad$ $T_{out\ water} = 440°C$

$C_C = 26.48\ m^3/hr$

$= 26.48 / 3600$ s

$= 0.00735\ m^3/s \times 947.9\ kg/m^3 = 6.97\ kg/s$

$C_C = 6.97 \times 4182 = 29158.28\ W/C$

$q = 29158.28\ (440 - 254.2) = 5.41 \times 10^6$ W

$C_h = \frac{q}{\Delta T} = \frac{5.41 \times 10^6}{541.1-423.3} = 45925.29\ W/C$

η NTU= $\frac{q}{C_{\min}(\overline{T}-\overline{t})}$

$\overline{T} = \frac{541.1+423.3}{2} = 482.2$ $\quad$ $\overline{t} = \frac{254.4+440}{2} = 347.2$

η NTU= $\frac{5.41 \times 10^6}{29158.28\ (482.2-347.2)} = 1.374$

NTU = $\frac{tanh^{-1}(\eta \text{ NTU} \times \frac{\sqrt{1+C_r^2}}{2}}{\frac{\sqrt{1+C_r^2}}{2}}$

$C_r = \frac{29158.28}{45925.29} = 0.635$

NTU = $\frac{tanh^{-1}(1.374 \times \frac{\sqrt{1+0.635}}{2}}{\frac{\sqrt{1+0.635}}{2}}$

NTU = 1.92

η NTU= 1.374

$\eta = \frac{1.374}{NTU} = \frac{1.374}{1.92}$

η = 71.5%

CHAPTER-6: Equipment Design

6.1–Economizer Design

1) <u>Outlet Temperature of Gases:</u>

SO_2 Concentration = 6.11 ppm

Partial pressure of vapors in Gases = 13.2 kPa

- Outlet Temperature by Graph is 137℃

T_1 = 70.45℃
T_2 = 29.1℃

LMTD = $\frac{70.45-29.1}{ln\frac{70.45}{29.1}}$

= **46.77**℃

2) <u>Updated values/readings:</u>

Standard deviation = 3

- <u>WATER</u>
Previous data
Density = $\frac{mass\ flowrate}{volumertic\ flowrate}$

Volumetric flowrate = $\frac{28.5}{930}$

Volumetric flowrate = 0.0306 $\frac{m^3}{s}$

Present data
Volumetric flowrate = A × v
Volumetric flowrate = $\frac{\pi\times d^2}{4}\times N\times v$
N = 122.7 × $\dot{V}$
120 = 122.7× $\dot{V}$
$\dot{V}$= 0.511 m/s

3) <u>Reynolds number:</u>

$$Re = \frac{\rho v l}{\mu} = \frac{vl}{\nu}$$

$$Re = 5.45 \times 10^4$$

4) <u>Pradl number</u>:

$$Pr = \frac{C_P \times \mu}{k}$$

$$Pr = 1.39$$

5) <u>Nusselt number</u>:

$$Nu = \frac{hl}{k}$$

$$Nu = 273$$

$$h = 7425.6 \, \frac{W}{m^2K}$$

6) <u>Shell side calculations</u>:

$D_s = 1.91$ m $\qquad\qquad A_s = 2.85$ m^2

$$A_{gas} = A_{total} - A_{tubes}$$

$$= 2.85 - 0.11$$

$$= 2.74 \text{ m}^2$$

Volumetric flowrate = $V = \frac{mass\ flowrate}{density} = \frac{m}{\rho}$

$$V = \frac{59.7}{0.854}$$

$$= 69.91 \, {m^3}/{s}$$

$$V = \frac{69.01}{2.74}$$

$$= 25.\,47 \, {m}/{s}$$

$$V_{max} = 38.32 \, {m}/{s}$$

- Reynolds number = 4.05×10^4
- Prandl number = 0.7358
- Nussel number = 748.83
- $h_s = 686.08 \, \frac{W}{m^2K}$

7) <u>Carbon steel</u>:

$$\frac{1}{U} = 3.29 \times 10^{-3}$$

$U = 521.95 \frac{W}{m^2K}$

$Q = U \times A \times LMTD$

$Q = U\times 2\times \boldsymbol{\pi}\times L \times d_o\times LMTD$

$= 5919$ kJ/s

$L= 9.73$ m

8) <u>Heat Exchanger</u>:

Surface Area of Heat Exchanger = $\boldsymbol{\pi}\times L\times d$

$= \pi\times 9.73\times 1.91$

$= 58.38\ m^2$

Baffles space = 1.91×0.4

$= 0.764$ m

Number of Baffles = $\frac{9.73}{0.764}$

$= 12.73 \cong 12$

6.2–Superheater Design:

- D_i = Wall thickness = 3.6 mm
  D_o = Outer diameter =0.0417 m
  1) <u>Superheater Inlet:</u>
     Temperature = 270℃
     Pressure = 545 barg

  2) <u>Tube side:</u>

$Q_{shell} = m\ (H_{out} – H_{in})$

$= 20.8\ (3193.27 – 2789)$

$= 8408$ kJ/s

$Q_{tubes}= 8.41$ MW

$LMTD = \frac{253.3-105}{ln\frac{253.3}{105}}$

LMTD = 168.4℃

$\text{Density} = \frac{mass\ flowrate}{volumertic\ flowrate}$

$20.26 = \frac{20.8}{V}$

V = 1.026 m^3/s

Volumetric flowrate = A×v

Volumetric flowrate = $\frac{\pi \times d^2}{4} \times N \times v$

N = 90

3) Reynolds Number:

$$Re = \frac{\rho v l}{\mu} = \frac{vl}{\nu}$$

$$Re = 3.64 \times 10^5$$

4) Prandl Number:

$$Pr = \frac{C_P \times \mu}{k}$$

$$Pr = 1.0779$$

5) Nusselt Number:

$$Nu = \frac{hl}{k}$$

$$Nu = 797.48$$

$$h = 1150.3 \frac{W}{m^2 K}$$

6) Shell side calculations:

D = 1.96

Cross sectional Area = 3.017 m^2

Area for Flue gases = 3.017 – $[(\frac{\pi}{4} \times 0.0417^2) \times 90]$

= 2.894 m^2

Volumetric flowrate = V = $\frac{mass\ flowrate}{density} = \frac{m}{\rho}$

$$V = \frac{59.8}{0.7321}$$

= 81.55 m^3/s

Standard deviation = 0.11676

Volumetric flowrate = V = $\frac{mass\ flowrate}{density} = \frac{m}{\rho}$

$$V = \frac{81.55}{2.894}$$

$= 28.1776$ m/s

$V_{max} = 43.83$ m/s

$D_e = 0.3745$ m

Reynolds Number

$$Re = \frac{\rho v l}{\mu} = \frac{vl}{\nu}$$

$$Re = 3.206 \times 10^5$$

Prandl Number:

$$Pr = \frac{C_P \times \mu}{k}$$

$$Pr = 0.7627$$

Nusselt Number:

$$Nu = \frac{hl}{k}$$

$$Nu = 657.44$$

$$h = 898.02 \frac{W}{m^2K}$$

7) <u>Heat Exchanger:</u>

$$k = 39.1 \frac{W}{mK}$$

$$\frac{1}{U} = 2.1131 \times 10^{-3} \frac{m^2K}{W}$$

$$U = 473.21 \frac{W}{m^2K}$$

$$Q = U \times A \times LMTD$$

$$Q = U \times 2 \times \boldsymbol{\pi} \times L \times d_o \times LMTD$$

$$L = 8.95 \text{ m}$$

Surface Area $= \boldsymbol{\pi} \times d \times L$

$= \boldsymbol{\pi} \times 8.95 \times 1.96$

$= 55.11$ m^2

$D_s = 1.96$ m

$B = 0.784$ m

Number of Baffles = 11

Baffle cut = 25-35 %

6.3–Evaporator Design:

In case of vaporization of water by using flue gas the value of U_d lies in the range of 40- 60 kW/m^2K

By hit & trial method,

U_d = 50 kW/m^2K

Area required = A = Q/UDΔT

T_{inlet} = 243°C = 516K T_{outlet} = 270°C= 543K

Steam Temperature = 258.1°C = 531.1K

LMTD = 27K

$Q = C_P\,\Delta T = \ m\widehat{H_{in}}$

$C_{P\ water}$ = 4.2 kJ/kgK $C_{P\ steam}$ = 1.03 kJ/kgK

H_{in} = 4.2 × (516 - 25) = 2.0622 × 10^3 kJ/kg

Q_{in} = 28.9 kg/s × 2.0622 × 10^3 = 59597.58 kJ/s

H_{out} = 1.03 × (516 - 25) = 533.54 kJ/kg

Q_{out} = 28.9 kg/s × 533.54= 15419.3 kJ/s

ΔQ = 59597.58 - 15419.3 = 44178.274 kJ/s

Heat of vaporization = 48134.5 kJ/s

Total Q = 48134.5 + 44178.274 = 92312.8 kJ/s

A = $\frac{92312.8}{50\times 27}$

= 68.37 m^2

Consider 16 BWG, ¾ inch OD, tube of 33.56 ft long tube

$\frac{outside\ surface\ area}{ft\ tube\ length} = \frac{\pi DL}{12}$

$$= \frac{3.14\times\frac{3}{4}\times 33.56}{12\times 16} = 0.412\ \frac{ft^2}{ft} = 1.352\ \frac{m^2}{m}$$

Tube wall thickness = 0.056 inch = 0.001422 m

ID of tube = 0.052 ft = 0.01585 m

No of tube required = $N_t = \frac{Area}{length} \times \frac{Area}{ft}$ = 129.45

Tube side: water

Area for flow = $\frac{\pi \times ID \times 2N_t}{4}$

$= \frac{3.142 \times 0.01585 \times 2 \times 129.25}{4}$

$a_t = 3.22\ m^2$

$G_t = W/\ a_t$

$= 26.29\ /\ 3.22$

$= 8.1645\ g/m^2hr$

$T_{avg} = 256.5℃$

$\mu = 0.000282\ kg/ms$

$Re = \frac{DGt}{\mu} = \frac{0.01585 \times 8.1645}{0.000282} = 458.89$

$\frac{L}{D} = 445$

From graph

j_h=0.04

$h_i = j_h \times \frac{K}{D} \times (\frac{C_p\mu}{K})^{\frac{1}{3}} \times Q_t$

Q_t = 1 assume

$h_i = 0.04 \times \frac{0.04206}{0.01585} = 0.106\ W/\ m^2K$

$h_{io} = 0.106 \times \frac{0.01585}{0.1987} = 0.00855\ W/\ m^2K$

Shell side: steam:

$h_o = 680\ W/\ m^2K$

as overall coefficient for pre heating:

$$U_c = U_p = h_{io} \times h_o(h_{io} + h_o)$$

$$= 0.6659\ W/\ m^2K$$

Surface area required for pre heating:

$$A_p = \frac{Q}{U_P \Delta T}$$

$$= \frac{44178.274}{0.6659 \times 27}$$

$$= 2457.17\ m^2$$

Vaporization:

At tube side vaporization is at 258.1℃ the heat transfer coffiecient for boiling water is 50 kW/m^2K

$h_{io} = 50\ kW/m^2K$

Overall coefficient for vaporization:

$$U_c = h_{io} \times h_o(h_{io} + h_o)$$

$$= 48.9\ W/m^2K$$

Dirt factor:

$$R_d = \frac{U_c - U_d}{U_c \times U_d} = \frac{50-48.9}{48.9\times 50} = 0.000449$$

Pressure Drop tube side (pre heating):

Re = 458.89

f = 0.0045

Length of pre heating zone = $\acute{L} = \frac{L\times A_p}{Ac} = \frac{10.23\times 2457.17}{2509.53}$

$$= 10.01\ \text{m}$$

Specific gravity = s = 1.46

Pressure Drop:

$$\Delta P = \frac{f\times {G_t}^2\times L_n}{5.22\times 10^2\times D_e\times s} = \frac{0.0045\times 8.1645^2\times 10.23}{5.22\times 10^2\times 0.02308\times 1.46} =$$

$$= \quad 0.371\ \text{psi}$$

Diameter of Shell:

Shell dia. can be found by:

$$N_t = \frac{\left\{(D_s - K_1)\left(\frac{\pi}{4}\right) + K_2\right\} - P_t(D_s - K_1)[K_3\times n + K_4]}{1.233\times P_t^2}$$

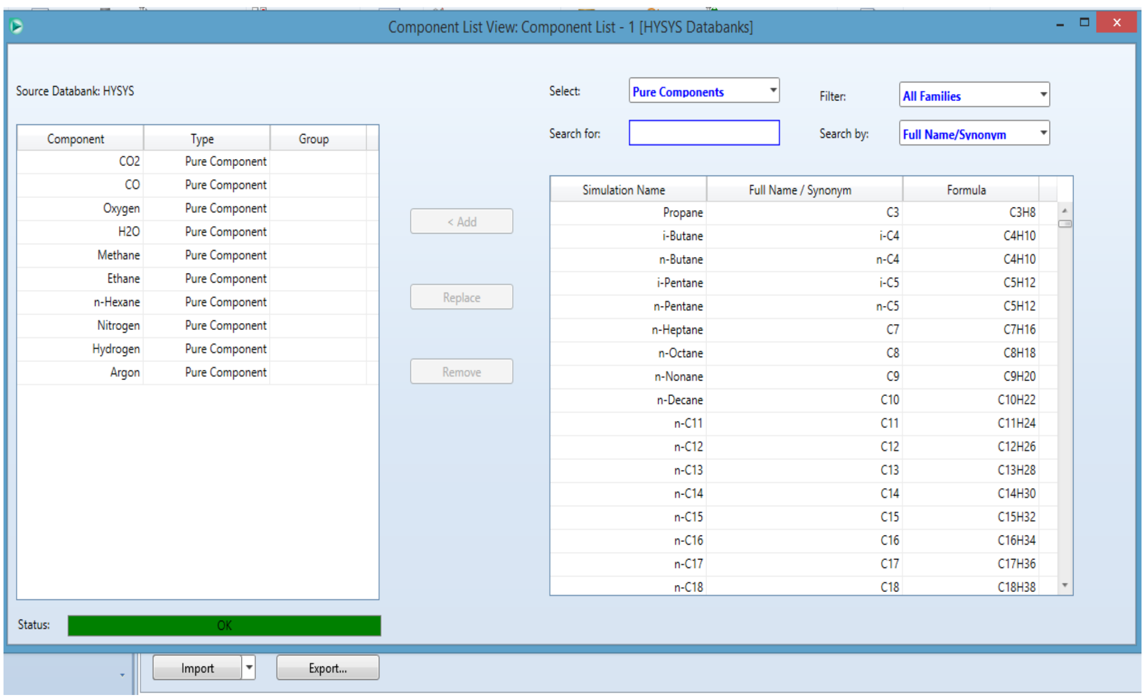
Ds = Dia of shell K_1= 1.08 K_2 = -0.9 $K_3 = 0.9$ $K_4 = -0.8$ $N_t = 129$

P_t = 1.25D = 0.0679m n= number of tube side passes

Ds = 3.28 m

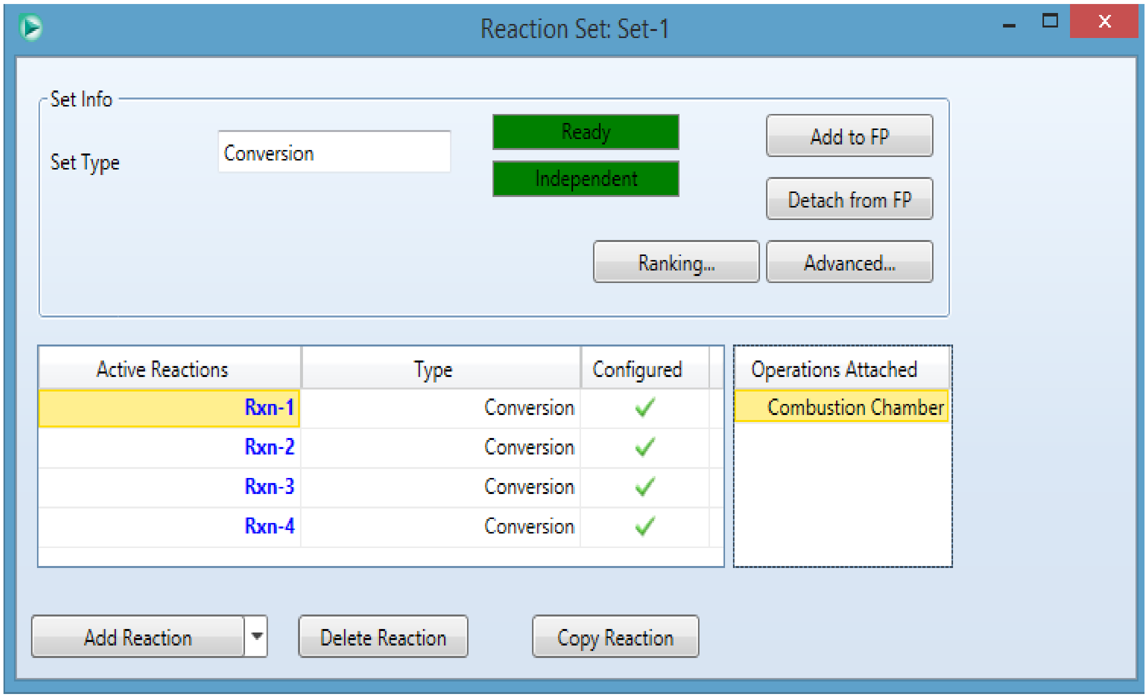
CHAPTER-7: Process Simulation on Aspen Hysys

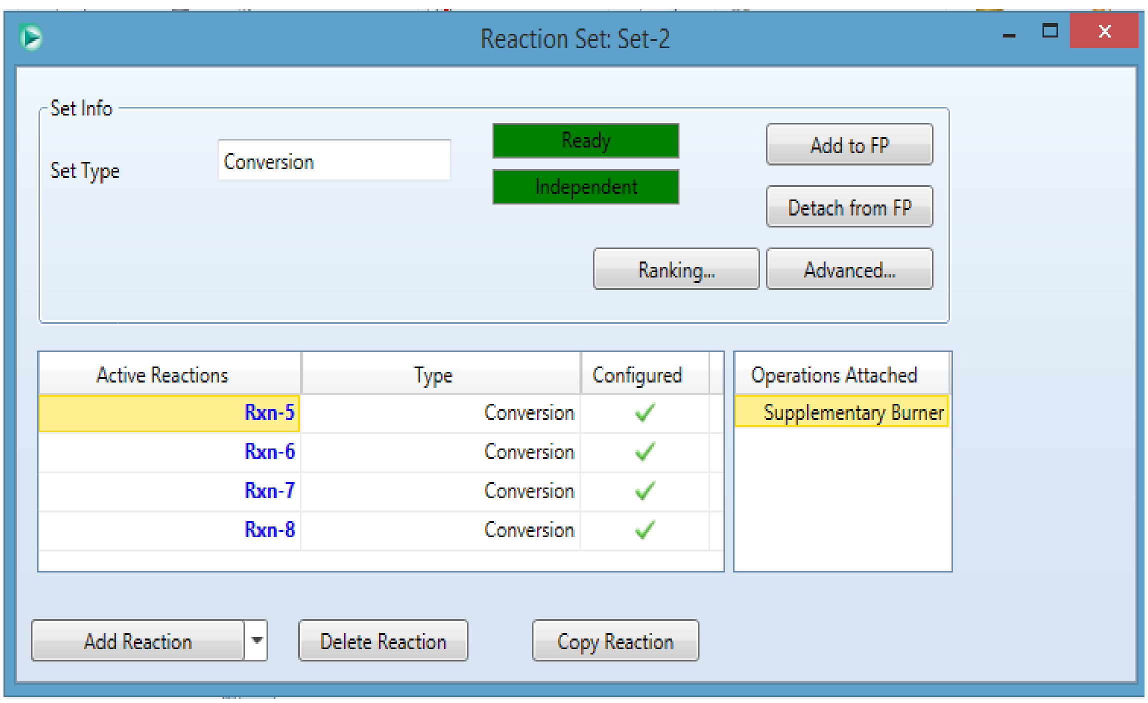
The simulation of whole co-generation cycle was made on Aspen Hysys. The required components were selected from hysys properties database are listed here in figure:



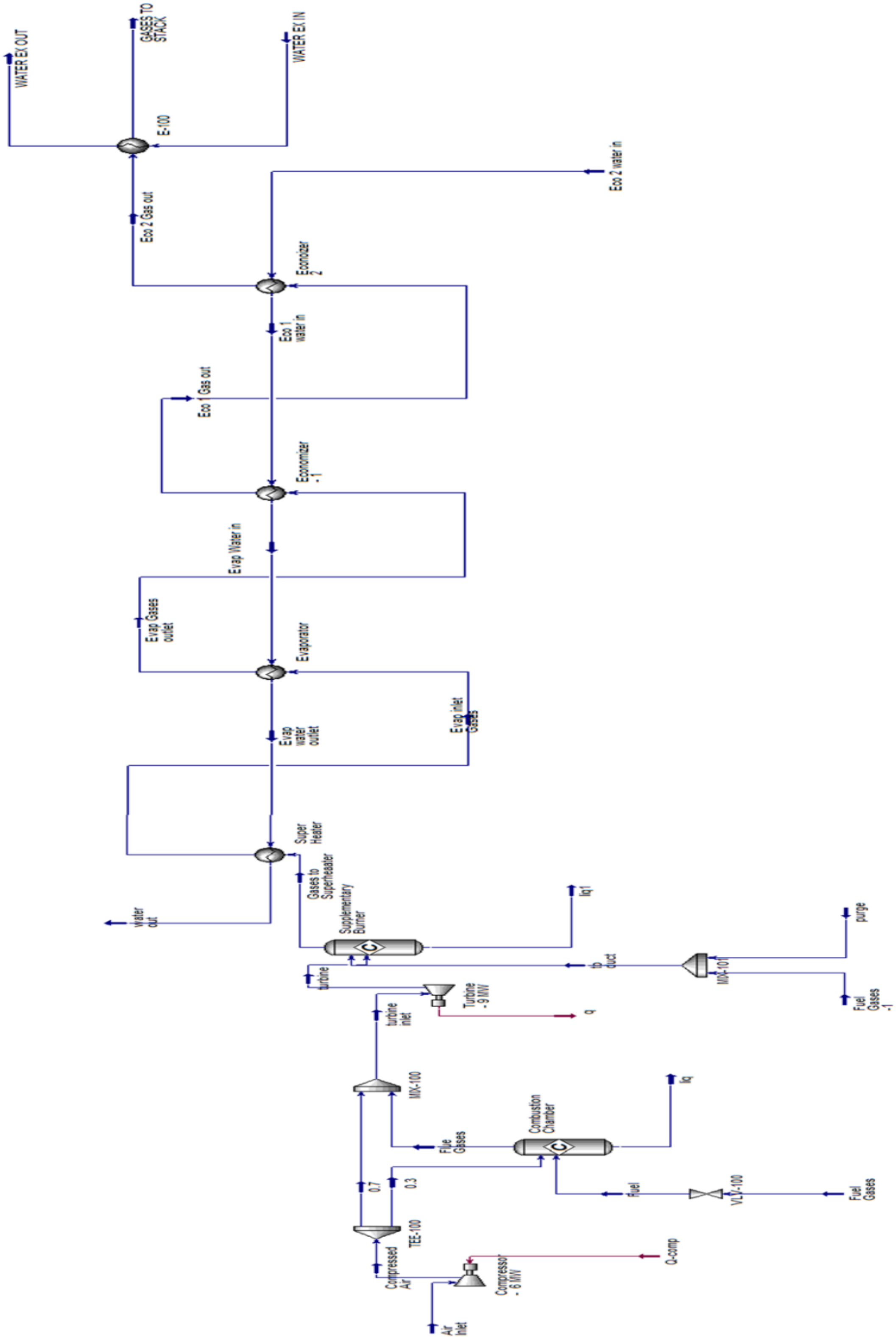
Reactions:

Two different reactions sets are used, each for a furnace. There are four active components in each of the furnace hence, four reactions in each set. Combustible components include: hydrogen, methane, ethane, hexane.


Reaction Set: Set-1
Set Info
Set Type
Conversion
Ready
Independent
Add to FP
Detach from FP
Ranking...
Advanced...
Active Reactions
Type
Configured
Operations Attached
Rxn-1
Conversion
Combustion Chamber
Rxn-2
Conversion
Rxn-3
Conversion
Rxn-4
Conversion
Add Reaction
Delete Reaction
Copy Reaction


Reaction Set: Set-2
Set Info
Set Type
Conversion
Ready
Independent
Add to FP
Detach from FP
Ranking...
Advanced...
Active Reactions
Type
Configured
Operations Attached
Rxn-5
Conversion
Supplementary Burner
Rxn-6
Conversion
Rxn-7
Conversion
Rxn-8
Conversion
Add Reaction
Delete Reaction
Copy Reaction

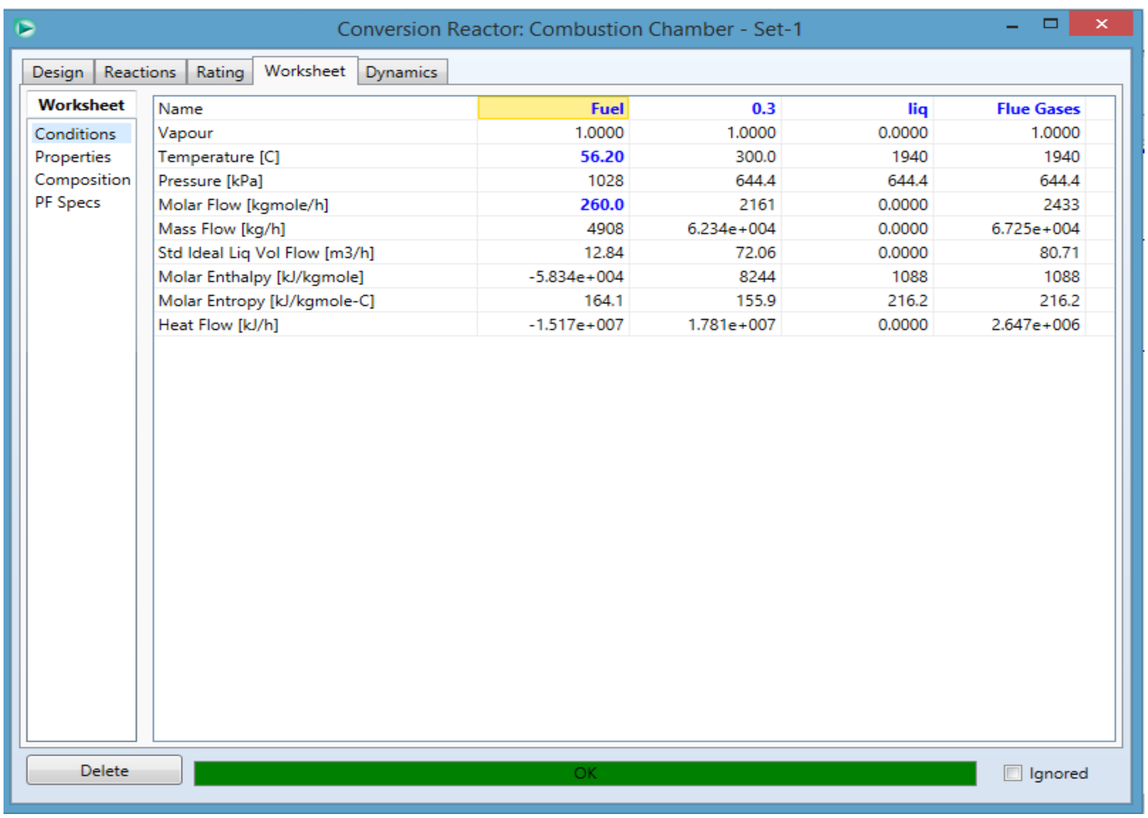
Flowsheet:



Process Brief Overview:

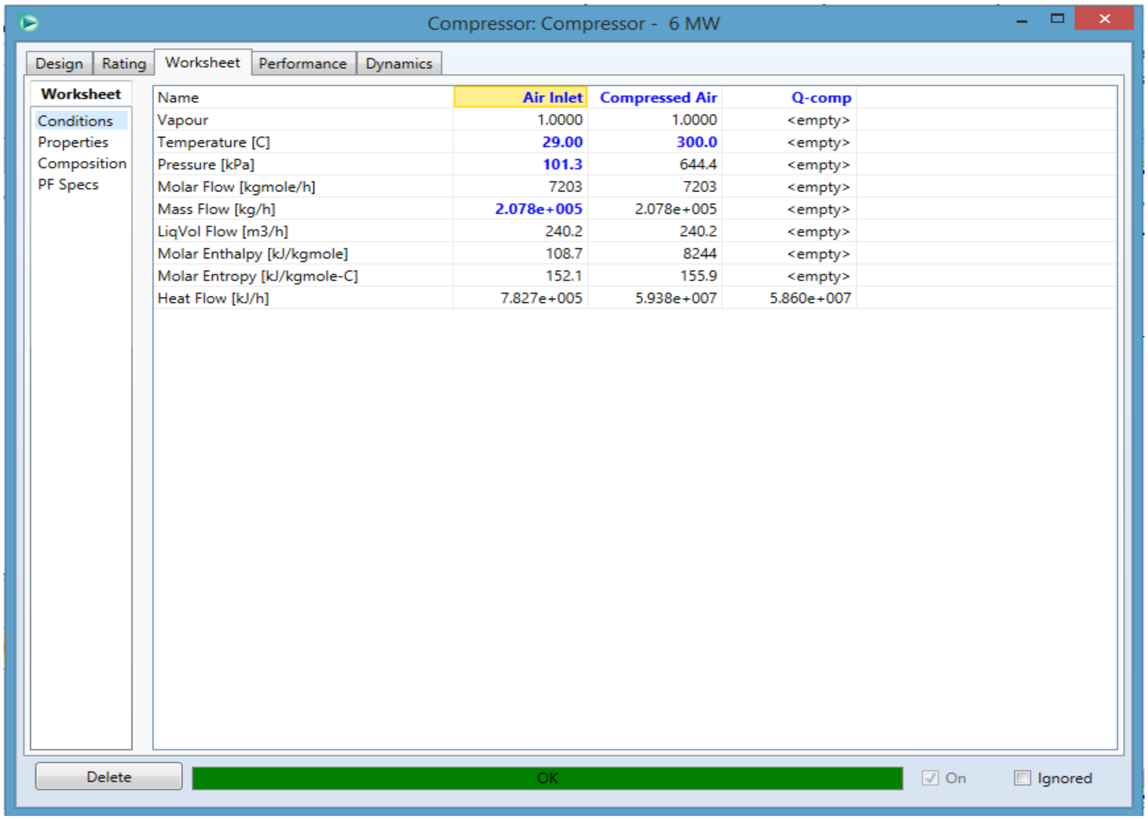
The process starts with the compression of air to 6 bars by a axial flow compressor, the compressed air's 30% goes to combustion chamber where it meets fuel gases and after mixing, they are ignited here. The hot goes leave the chamber at 1800°C, which is too hot for Gas Turbine to handle, so the remaining 70% of air is added into it, from compressor, to bring its temperature to 900°C. The flue gases finally go to turbine where they rotate the blades connected on a single shaft, which is eventually used to run the generator. The flue gases after being expanded in turbine leave it at 450°C, which is finally taken to a 'Heat Recovery and Steam Generator'. Here a series of equipment including, 'Superheater, Evaporator, and two Economizers' extract the thermal energy from the gases. Making the whole process energy efficient, as the heat recovered is used up in steam generation.

Main Combustion Chamber:



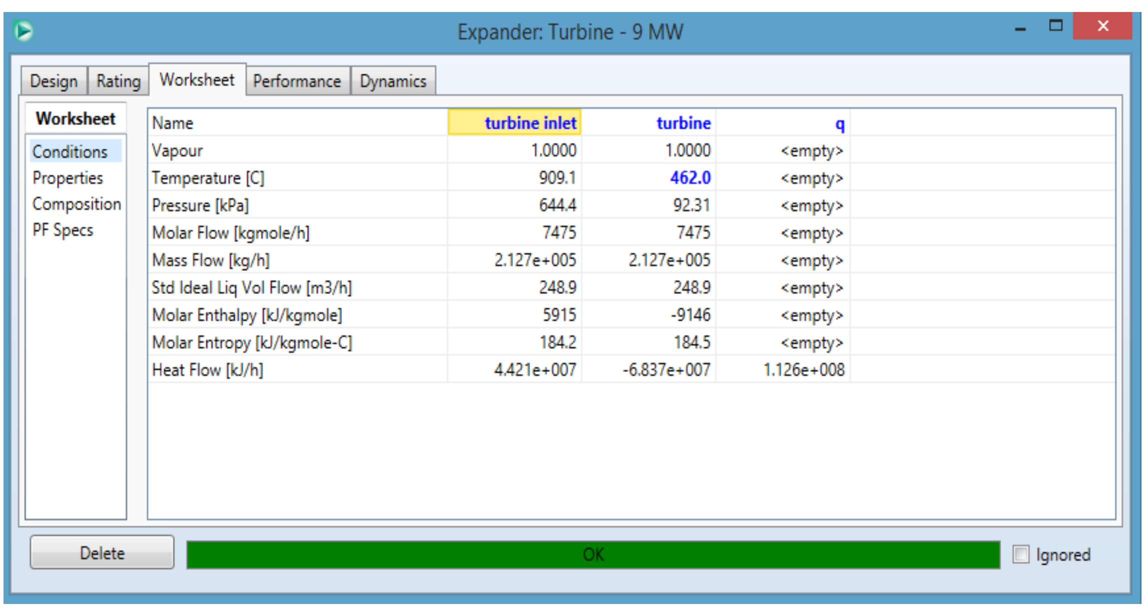
Name	Fuel	0.3	liq	Flue Gases
Vapour	1.0000	1.0000	0.0000	1.0000
Temperature [C]	56.20	300.0	1940	1940
Pressure [kPa]	1028	644.4	644.4	644.4
Molar Flow [kgmole/h]	260.0	2161	0.0000	2433
Mass Flow [kg/h]	4908	6.234e+004	0.0000	6.725e+004
Std Ideal Liq Vol Flow [m3/h]	12.84	72.06	0.0000	80.71
Molar Enthalpy [kJ/kgmole]	-5.834e+004	8244	1088	1088
Molar Entropy [kJ/kgmole-C]	164.1	155.9	216.2	216.2
Heat Flow [kJ/h]	-1.517e+007	1.781e+007	0.0000	2.647e+006

Compressor:



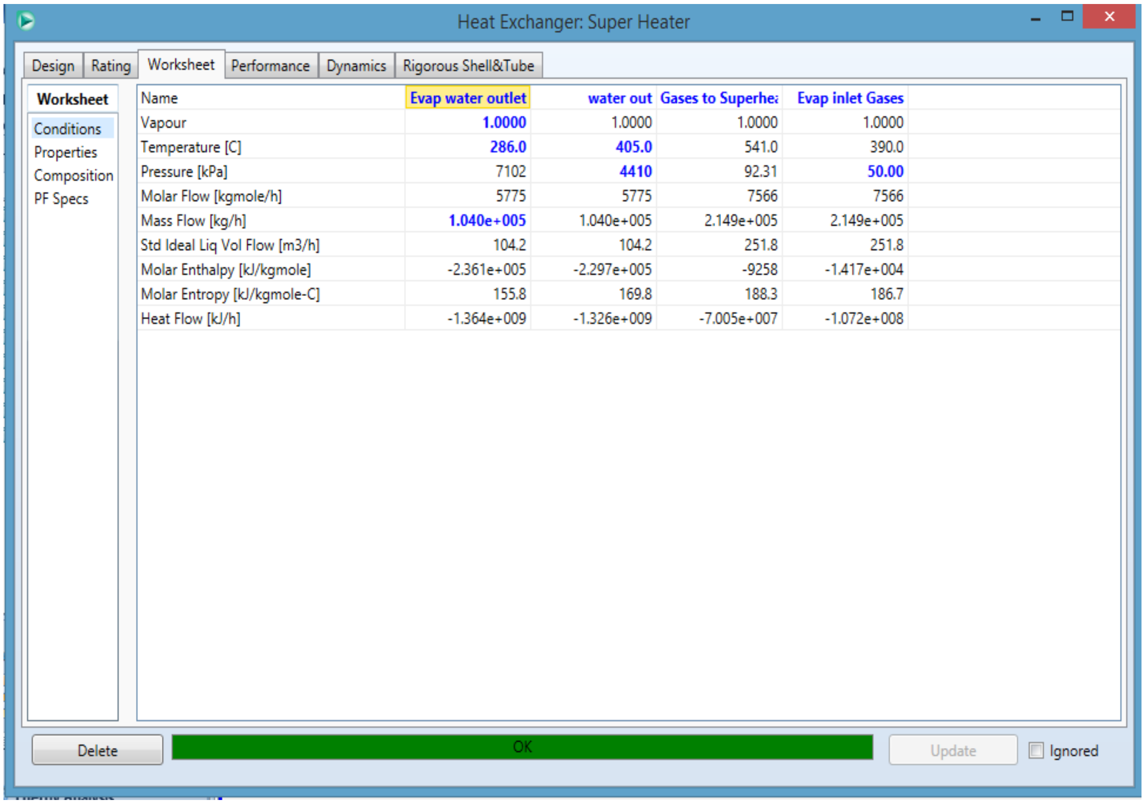
Name	Air Inlet	Compressed Air	Q-comp
Vapour	1.0000	1.0000	<empty>
Temperature [C]	29.00	300.0	<empty>
Pressure [kPa]	101.3	644.4	<empty>
Molar Flow [kgmole/h]	7203	7203	<empty>
Mass Flow [kg/h]	2.078e+005	2.078e+005	<empty>
LiqVol Flow [m3/h]	240.2	240.2	<empty>
Molar Enthalpy [kJ/kgmole]	108.7	8244	<empty>
Molar Entropy [kJ/kgmole-C]	152.1	155.9	<empty>
Heat Flow [kJ/h]	7.827e+005	5.938e+007	5.860e+007

Turbine:



Name	turbine inlet	turbine	q
Vapour	1.0000	1.0000	<empty>
Temperature [C]	909.1	462.0	<empty>
Pressure [kPa]	644.4	92.31	<empty>
Molar Flow [kgmole/h]	7475	7475	<empty>
Mass Flow [kg/h]	2.127e+005	2.127e+005	<empty>
Std Ideal Liq Vol Flow [m3/h]	248.9	248.9	<empty>
Molar Enthalpy [kJ/kgmole]	5915	-9146	<empty>
Molar Entropy [kJ/kgmole-C]	184.2	184.5	<empty>
Heat Flow [kJ/h]	4.421e+007	-6.837e+007	1.126e+008

Super Heater:



Name	Evap water outlet	water out	Gases to Superhe:	Evap inlet Gases
Vapour	1.0000	1.0000	1.0000	1.0000
Temperature [C]	286.0	405.0	541.0	390.0
Pressure [kPa]	7102	4410	92.31	50.00
Molar Flow [kgmole/h]	5775	5775	7566	7566
Mass Flow [kg/h]	1.040e+005	1.040e+005	2.149e+005	2.149e+005
Std Ideal Liq Vol Flow [m3/h]	104.2	104.2	251.8	251.8
Molar Enthalpy [kJ/kgmole]	-2.361e+005	-2.297e+005	-9258	-1.417e+004
Molar Entropy [kJ/kgmole-C]	155.8	169.8	188.3	186.7
Heat Flow [kJ/h]	-1.364e+009	-1.326e+009	-7.005e+007	-1.072e+008

CHAPTER-8: Instrumentation & Process Controls

8.1–Automatic Process Control

Introduction:
Automatic process control is a system that is applied to different processes in order to control process conditions, parameters and variables and to keep them at the required values. Automatic process control can maintain temperature, pressure, process variables, flow, compositions and some other parameters at the desired value.
For any process, control is very important for many reasons. Based on the industrial experiences the most important reasons for applying a process control system include:

- It can prevent accidents or any kind of damage to plant or any personal. It keeps the process conditions under control and within the safety limits.
- It can protect environment by reducing waste and minimizing emissions.
- Safety is the most important consideration while applying control system so it ensures that the plant is always working under safety limits.
- It can maintain the product quality, colour, composition and purity etc.
- It can reduce the cost of production and ensure smooth operation.

So the automatic control systems are applied to maintain product quality, increase production rate, reduce the need of labour and at the same time provide the safe environment.

Drawbacks of poorly performing control loops:
Operability and profitability of industrial plants can be effected by the control loop. Considering the importance of control loops, one would expect that they always perform at their peak, but this is not the case. In fact, several studies have shown that roughly one third of industrial control loops perform poorly. Poorly performing control loops can make a plant difficult to operate and may have several costly side-effects, including:

1. Reduced production rate
2. Increased emissions
3. Lower efficiency
4. Plant trips following process upsets
5. Poor product quality
6. Slower start up and transition times
7. More off-spec product or rework
8. Premature equipment wear

For these reasons, control loop performance should always be kept at the highest possible level. This is achieved through continuously monitoring loop performance and taking the appropriate corrective actions when sub-optimal performance is detected.
To effectively manage, improve, or sustain control loop performance, it is important to monitor how well loops perform. Loop performance monitoring can provide valuable feedback on the success of control optimization projects; it helps maintaining a high standard of loop performance in the long run; and it can be used to pinpoint offending control loops for corrective action.

Basic Components:
There are three basic components of any control system

- Sensor-Transmitter
- Controller
- Final Control Element

Sensor and Transmitter:
A sensor element measures a process variable: flow rate, temperature, pressure, level, pH, density, composition, etc. Much of the time, the measurement is inferred from a second variable: flow and level are often computed from pressure measurements, composition from temperature measurements.
These are the primary and secondary devices respectively.
Sensor:
Sensor also called primary element. Its purpose is to detect changes in the environment and provide the resultant output. It is a device that is used to measure the process variables for example temperature, pressure, flow rate etc.
Different sensors for different purposes include:

Measurement	Sensors
Temperature	Thermocouple, Thermistors, RTD's, IC
Pressure	Fibre optic, elastic liquid based manometers, Vacuum, electronic.
Flow	Differential pressure, Electromagnetic, Thermal mass, Positional Displacement.
Level	Thermal displacement, ultrasonic radio frequency, Differential pressure.

Sensor Selection:
Sensors/ transmitters are selected based upon certain features including:

- **Range:** It is the measurement limit of the sensor
- **Calibration:** It is the extent to which the sensor measures the correct value.
- **Accuracy:** It is how precisely a sensor measures a value.
- **Resolution:** It is the smallest change in value that is measured by the sensor.

- **Response:** It is the time that a sensor takes to reach the true value when there occurs a change in the value which it is measuring.

- **Sensitivity:** It is one of the most important parameters while selecting a sensor. It is the change in output of the sensor per unit change in the value it is measuring.

- **Cost:** It depends on how much expensive sensor can one afford. It is increased with increased accuracy, better precision, longer range and longer life.

The sensor that is being used in this system for temperature measurement is thermocouple.

Thermocouple:
Thermocouple is a temperature measuring device. Thermocouples contain two legs of wires that are made up of different materials. Both of these wires are weld together on a single end, this will create a junction. This is the junction where temperature is measured. When the junction experiences a change in temperature, a voltage is created.

Transducers:
A transducer is a device that receives a signal and retransmits it in a different form. For example, we've discussed me /P transducers that convert a current signal to pneumatic form. It is also called convertor. After a value is measured by the sensor, it is the received by the transducer. Transducer is a device that receive a signal from the sensor and retransmit in a different form that is readable by the controller.
There are many different types of transducers including:

- **Transducer I/P:** It will convert electric signals in to pneumatic ones. Input may be 4 to 20 mA and output is in 3 to 15 psig.
- **Signals:**
  There are three basic types of singles that are used in any industry:
- **Pneumatic Signal:** Pneumatic signal is also called air pressure. It normally ranges between 3 - 15 psig. It is usually represented in diagram as,

- **Electric Signal:** It is also the most common type of signal used. It ranges between 4 – 20 mA, 10 – 50 mA, 1 – 5 V or 0 – 10 V etc. It is usually represented as dash lines. –
  – – – – – –

Digital signal:
Third type is digital, or discrete signal. It is normally comprised of 0 & 1 combinations.

Controller:
Controller is a device that takes an input from the transmitter and perform accordingly to maintain the controlled variable at the set point. In this system we are using a feedback controller.

Feed Back Controller:
A feedback controller is a type of controller that takes input from the system's outlet. It compensates for all the disturbances due to which the controlled variable deviates from the set point. A feedback controller decide what to do to maintain the controlled variable at the set point. (Smith, 2005). A feedback controller comes to a decision by solving an equation based upon the difference between controlled variable and the set point.

PID Controller:
It is a proportional–integral–derivative controller (PID controller). A feedback control loop mechanism is used in industrial control systems. It calculates the error value continuously as the difference between the set point value and the controlled variable. The purpose of the controller is to minimize the error by controlling the final control element that is the control valve. It will either close or open the valve accordingly to adjust the value. (Smith, 2005)

Equation:
This is the equation for PID controller. The first term shows the value of proportional action, second is of integral which completely eliminates the offset but the oscillations are increased. So, the third term derivative is added, which reduces the number of oscillations.

Purpose of selection:
The PID is selected because temperature is inherently slow variable, and it takes time to develop it. So, PID is suitable for this as this will reduce the noise and the integral will remove or reduce the offset.

So the main purpose of PID is to give fast response, remove offset and less oscillations. The response is made faster by proportional part. If gain is increased the response is faster. The offset is removed by integral part. And the derivative action will control the over shoot and unnecessary oscillations.

Final Control Element:
This is the third main component of the control system. It takes an input from the controller and adjust accordingly to keep the controlled variable maintained at set point.

CHAPTER-9: HAZOP Analysis

The expression "HAZOP Analysis" means “risk and operability” study. It is deliberate and organized strategy for analysing an officially working framework or a framework which intended to be active in future. It is a subjective examination of the framework which is finished by a group of qualified specialists to discover the areas in framework which have high hazard potential towards gear and work. It is likewise used to turn away the conceivable unfriendly impacts to process productivity, which may happen because of the dangers at the specific areas. The investigation is subjective and furthermore distinguishes deviations which may possibly bring about process wasteful aspects. So this is operability examine not to be mistaken for "Hazard Analysis" which quantitatively measures the dangers related with the procedure.

Procedure:

Certain words are used in identification of steps used for carrying out the hazard and operability analysis (HAZOP). These words along with their brief definitions are given below:

- Intention:

This defines how a particular part of the process was intended to operate i.e. the intention of the designer.

- Deviations:

These are departures from the designer's intention. These deviations are detected by the systematic application of the guide words.

- Causes:

Here possible reasons for the deviation are considered answers to the questions like how and why are determined. Only the deviation resulting from a realistic cause is treated as meaningful.

- Consequences:

The results of a meaningful deviation (now meaningful can be large deviations for some processes and small for processing requiring critical control).

- Hazards:

Consequences that can potentially cause damage (loss) or injury.

Since an adiabatic pre-reformer is a fixed bed reactor where gases (natural gas + steam) are introduced from top and leave the reactor from bottom. Now looking at various parameters which can deviate various cases are considered.

Case 1:
Intention: The desired temperature for steam generated by superheated is 400℃
Deviation: The temperature goes beyond 400℃.
Causes: One of the reason for the shoot in steam temperature is malfunctioning of turbine. It might be reducing the load from turbine, which consequently cause more rich flue gases in thermal energy so they call imbalance the whole process going in "Heat Recovery and Steam Generator". Eventually giving steam at higher temperature then the desired one.
Consequences: This can be very costly, as the steam is being used for steam reforming, and variation in temperature will reduce the yield of Hydrogen. It will be the result shit of reactions equilibrium to the left. The end result of this all would be reduced ammonia yield and finally lesser urea production.
Hazard: The stated problem can be solved by the addition of "desuperheater" which mixes Boiler Feed Water with the steam generated from the superheater to bring it to the optimum point.

Case 2:
Intention: Maintaining the temperature of flue gases at the Superheater.
Deviation: Temperature goes beyond or below 500^{O}C.
Causes: The reason might be the variation of load at gas turbine or furnace malfunctioning, which might lead to higher temperatures at superheater, this can again take away the steam from the standard set for it.
Consequences: The occurrence of this even can damage the subsequent process taking place by the steam generated. Eventually, the cost of operation of the whole process will be increased.

Safety of the Personnel:
Prevention:
The best assurance of the wellbeing of the individual is to avoid the event of any unsafe circumstances. Under typical conditions, there will be no gases, vapor, fluid or chemicals released in the environment.

At the point when a break is created, it ought to be ceased or repaired on the double. On the off chance that this ought not be conceivable, different means must be connected to forestall spreading, to counteract mischance to work force and to keep away from harm to hardware.

Health Hazards:
At the point when a man harmed by the outer contact, inward breath or oral infusion of the poisonous substance, the medical aid directions for that compound must be taken without a moment's delay. Consequently, it is most essential that all individuals know about these medical aid guidelines.

Protective equipment:
At the point when the defensive gear is important it must be utilized appropriately. The hardware chosen must be suited for the reason and the individual utilizing the defensive gear must be comfortable with it. All defensive hardware for the wellbeing of individual must be kept in great working conditions at record-breaking.

Eye protection:
Goggles and face shields will be given to every administrator. He should be comfortable with the area and the utilization of eye shower and security showers should be done. 87

Respiratory protection:
All gases other than air are destructive to people when breathed in adequate fixation. Dangerous gases might be named either suffocating or bothering.
Suffocating gases might bring about death by displacing air in lungs or by responding with O2 conveyed in the blood. e.g., H2S, CO2, Smoke. Disturbing gases might bring about harm or passing by suffocation as well as both inner and consuming (e.g. Cl2, SO2, HF).
To guard against the inhalation of harmful gases the operator should,

• Score a gas test certification showing the gas conditions of the atmosphere (vessel).

• Avoid standing on the side of an opening where escaping gas will be blown towards him.

• Provide ventilation.

• Wear the gas mask when required.

Foot Protection:
It is prescribed to utilize shoes of ideal cowhide with an implicit steel toe top. Whenever important, boot of unique substance safe material can be put on.

Head Protection:
Security must be forever utilized inside the plant, particularly repairing is being made and there is plausibility devices and materials mishap.

Skin and Hand protection:
Suits made of engineered elastic or other endorsed material might be worn when finish body insurance is essential, gloves made of manufactured elastic or other affirmed material ought to be worn to ensure the hands. In the event of extreme releases just legitimately secured individual ought to stay in the territory.

CHAPTER-10: Economic Analysis & Costing

1) **Cost of Duct:**

Cost = bare cost × Material factor × Pressure factor

= 120,000 ×1 × 1

= $120,000/-

2) **Cost of Superheater:**

Area = 55.1 m^2

Bare cost = $60,000

Pressure = 30-50 bar

Total Cost = 60,000×1.3×0.8

= $ 62,400 /-

3) **Evaporator:**

C = Bare cost = $20,000

n = Index = 0.53

Cc = C(S^n)

= (20,000) ($73^{0.53}$)

= $ 194,353 /-

4) **Economizer:**

Bore cost = $23,000

Total cost = 23,000 ×1 ×1

= $23,000

Two economizers of same capacity = $23,000 × 2

= $46,000/-

5) **Stainless Steel:**

Thermal n = 0.23

Turbine power = 9 MW

$Q_{in} = \frac{9}{0.23}$

= 39.13×10^3 MW

Cc = C(S^n)

= (560) $(39.13 \times 10^3)^{0.77}$

= $3,849,900 /-

6) **<u>Cost of Turbine:</u>**

Cc = (3000) ($9000^{0.5}$)

= **$284,605 /-**

7) **<u>Compressors:</u>**

Cc = (1920) ($5600^{0.8}$)

= **$1913,600 /-**

8) **<u>TOTAL PURCHASE COST OF EQUIPMENT</u>**

Total purchase cost of equipment = Compressor + Furnace + Turbine + duct+Superheater +Evaporator + Economizer

= $6470.85

9) **<u>Physical Plant Cost</u>**

Physical plant cost = PCE x 3.4

= **$22,000,890 /-**

10) **<u>Fixed Capital</u>**

Fixed cost = **$31,901,290 /-**

11) **<u>Variable Cost</u>**

Variable cost = **$ 1.08 x 10^3 /year**

12) **<u>Total Variable Cost</u>**

Total variable cost = (9396.075 x 10^6) + (108.16 x 10^3)

= **$9.396×$10^6$ /year**

13) **<u>Total Fixed Cost</u>**

Total variable cost =$6138244. 6

14) **<u>Direct Production Cost</u>**

Direct Production Cost = Fixed Cost + Variable Cost

= 6138244.64 + (9.396 x 10^9)

= **$9.4 x 10^6**

CHAPTER-11: Conclusions

The combined cycle is power and steam generation simultaneously. A gas turbine of 9MW was coupled with a generator and the flue gases leaving the turbine were sent to a Heat Recovery and Steam Generator to extract thermal energy from and they left the system at 140°C.

- The thermal efficiency of the whole unit is increased by more than 8% via installing another 'Economizer'. This was exchanging heat with BFW, a condensate from another Boiler. It will lower the temperature of the gases leaving HRSG at 140°C, which was initially 227°C. Thereby increasing the overall efficiency of the system.

- Wet compression and fogging lowers the inlet temperature of the gas, increasing the efficiency of the gas turbine. This has lowered the inlet temperature of the ambient air going in the compressor. The temperature is inversely proportional to the power generated by the gas turbine.

CHAPTER-12: Recommendations

12.1–Fogging:

Consequently, warm efficiencies are right now extremely appealing; with straightforward cycle efficiencies extending between 32 percent and 42 percent and consolidated cycle efficiencies achieving the 60 percent stamp. The compression procedure consumes as much as 66 percent of the aggregate work delivered by the gas turbine, and consequently any methods for lessening crafted by compression will improve the power yield of the gas turbine. The yield of a gas turbine diminishes by increment in encompassing temperature, ordinarily 3 to 3.5 % with 10 degree F ascend in temperature. (cheerful)

Coordinate delta fogging is a strategy for cooling where de-mineralized water is converted into a mist by methods for high-weight spouts operating at 1000 psi to 3000 psi. This mist at that point gives cooling when it dissipates noticeable all around channel pipe of the gas turbine. This procedure permits 100 percent adequacy regarding accomplishing 100 percent relative stickiness at the gas turbine gulf, and subsequently gives the most reduced temperature conceivable without refrigeration (the wet globule temperature). Coordinate high-weight delta fogging can likewise be utilized to make a compressor intercooling impact by permitting abundance haze into the compressor, along these lines boosting the power yield considerably.

12.2–Water Quality:

The significance of water quality can't be exaggerated, particularly when haze intercooling is wanted. Experience has demonstrated that demineralized water often requires additional treatment on the off chance that it is to be utilized for gas turbine gulf air cooling. Demineralization by and large evacuates ionic material (minerals that are broken down in the water in ion shape) however does not expel colloidal material. Surface waters, for example, waterway or lake-water, and even some well waters often contain silica in colloidal shape. As silica is a hard component, its essence can harm mist spouts and perhaps even compressor blading. The haze pump slips or water treatment plant ought to incorporate submicron water channels to expel the silica from the mist supply water.

Operating problems and solutions:

a) Due to the utilization of fast positive relocation pumps, unnecessary vibration was noted on the high-weight lines. The installation of pulsation dampers settled this issue. While pulsation dampers are currently constantly utilized, moderate speed pumps intrinsically diminish vibration and wear on pump valves and seals.
b) The water supply stagnated and permitted the development of microbes, which caused stopping of the high-weight fogging spouts and obstructing of the channels. The issue was settled by support plans for depleting the framework. This underlying installation additionally changed from polyvinyl chloride (PV C) water supply channeling to stainless steel (SS) funneling, which is presently a standard element in fogging frameworks offered.
c) Some haze spout complex disappointments were experienced conceivable by vibration caused via wind current. The issue was settled by fortifying the spout manifolds to the structure.
d) Currently, this installation uses its chillers only when economical (at specific circumstances of the day) and utilizations its fogging framework, which is more affordable to work amid off-top hours. This is a decent case of a crossover framework wherein both cooling advances are incorporated and permit an ideal solution.
e) Fogging frameworks shower atomized water into the GT channel air. Evaporative cooling framework consists of soaked media through which the GT bay wind current goes, to chill off by evaporation. Chiller framework is like aerate and cool chiller used to cool huge structures. In this framework, chilled water is coursed through a finned-tube curl put in the GT delta air way. This chills off the delta air, perhaps condensing a portion of its dampness, which is depleted away.

Measure of water splashed in channel air stream characterizes the sort of gulf fogging, specifically under shower and over shower. In under splash, air is cooled by dissipating mist, without beads entering the compressor. Be that as it may, in finished splash, extreme water beads enter the compressor and will influence its execution. Besides, certain gas turbine motors are unsatisfactory to overspray fogging, and a few producers don't prescribe overspray for GT.

An expansion in turbine limit was discovered when the compressor water wash framework was coincidentally left open at one of Dow Chemical's offices. This incited advance investigation into the procedure and brought about licenses issued to Dow Chemical on this innovation.

12.3–Wet compression:

In evaporative cooling, the air is cooled by bringing water into the air either by conventional media write evaporative coolers or by coordinate shower through spout exhibits. The reduction noticeable all-around temperature is restricted by the capacity of air to retain water. The outline objective was to include only the measure of water to soak the air before it enters the compressor. The vestige of water into the compressor was to be maintained a strategic distance from definitely. There has been an adjustment in this reasoning in the last 5-7 years.

The introduction of water into the compressor gulf brings about considerable picks up in turbine yield gave appropriate care is taken to guarantee uniform shower distribution, satisfactory bead sizes and monitoring of basic turbine parameters. This technique for abundance water splash into the air stream is often named as "wet compression", "over-shower", or "high-fogging".

A region of essential concern in the prior years for the application of fogging frameworks was the likelihood of blading erosion. With the little bead measurements being produced and utilization of demineralized water (DM), erosion identified with bay air fogging and haze intercooling has been demonstrated not to be an issue. There is no negative impact on the blading from an erosion standpoint when high-weight fogging is utilized, and in fact some prior examinations have shown that considerably higher bead sizes don't have any adverse erosion impact.

It is intriguing to take note of that compressor water injection has really been proposed to enhance execution of a dissolved hub stream compressor. This should be possible in light of the fact that the water injection expands the aggregate weight proportion, hence recuperating some erosion-related deterioration.

12.4–Impact of fogging on emissions:

There are a few investigations that have shown that NOx emissions can be decreased by fogging of the bay air. Trial chip away at a LM2500 unit showed NO_x reductions of roughly 33 ppm over the scope of motor forces with the addition of 4.5 gpm of water to the delta of the motor's compressor.

Warm NO_x is an exponential function of fire temperature. Hotspots in the fire are responsible for the greater part of NO_x production. The water vapor presented by bay fogging, having a particular warmth around four time that of air has extinguishing impact.

12.5–New air channels:

Fouling is caused by the adherence of particles to the airfoils and annulus surfaces. The adherence is caused by oil and water fogs the outcome is the development of material that causes expanded surface unpleasantness and to some degree changes the state of the air thwart particles that reason fouling are commonly littler than 2 to 10 micrometer. Smoke old fogs carbon and ocean salts re common cases. Stopping of turbine gap cooling sharp edges, caused by sub micron molecule advance harm from overheating. Air contamination can be fundamentally lessened however the utilization of productive filtration framework particularly those that diminish the amount of materials ingested.

Better channels are required in light of the fact that

a) Online cleaning only cleans the initial couple of stages. The reagent dissipates as the temperature and weight increments in the compressor consequently leaving the later stages still unclean.
b) Dirt particles broke down in the reagent can redeposit in later stages or can be conveyed to the hot section of the turbine. The last impact is more hazardous as sodium and potassium are enter fixings in hot corrosion of turbine.

Beforehand planned reasoning was that an all-around composed air filtration framework for a given application would perform well in about any environment with square with comes about. However now numerous organizations offer an extensive variety of filtration house for gas turbine to enable them to perform effectively in particular operating conditions. Measured construction implies custom and standard plan are accessible in self-cleaning, invert beat, hindrance up stream, cross stream, and downstream models. These units are utilized for quick and simple erection onsite. There is a tradeoff between weight drop cost, size, and filtration productivity.

As far as gas turbines channels incline is moving towards self-cleaning channels. Manufactured channels are being made to supplant more seasoned less effective cellulose fiber write media and often give better arrival of clean particles.

Diverse wellsprings of contaminants deliver particles of unfathomably unique size. Heartbeat channel lodging has additionally made strides. Most channels are accessible now with corrosion safe covering of zinc and stainless steel metals. Heartbeat channels presently have included quality and they have better cement forces to avert sidestep into air stream.

The 100 percent engineered media is a combination of three layers of channel medium. They perform four basic filtrations; course filtration; fine filtration; hydrophobicity and mechanical uprightness. Already HEPA were composed in a way that modifications were required in the current channel house. The new HEPA would now be able to be coordinated with the current equipment of the channel house.

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