Increasing Gas Turbine Efficiency by Inlet Air Chilling

By

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Certificate

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DEDICATION

TO

OUR PARENTS

without whom none of this would have been possible and for their support throughout our lives

AND TEACHERS

for inspiring us and supporting us throughout the entirety of the project

ACKNOWLEDGEMENT

It is a pleasure and a deep sense of indebtedness that we acknowledge the valuable help of our respected supervisor who have enriched the text by his generous contribution and patronage. We express our cordial gratitude to our supervisor **Dr. Zaib Jahan** for her encouragement, patience guidance, enthusiastic support, constructive suggestions and ever helping supervision.

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Author

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Nomenclature

- Cp Specific heat capacity at constant pressure
- Cv Specific heat capacity at constant volume
- P Pressure
- P Power
- T Temperature
- v volumetric flowrate
- n Number of moles
- R Universal gas constant
- ρ Mass density
- M Relative molecular mass
- H Enthalpy
- H Head
- m mass flowrate
- Q Heat flow
- Δ change in
- d Diameter
- A Area
- π pi
- Re Reynolds number
- μ Viscosity
- Pr Prandtl number
- k Thermal conductivity
- Nu Nusselt number
- Jh Chilton and Colburn factor
- h Heat transfer coefficient

- U Overall heat transfer coefficient
- l Length
- G Mass velocity
- t time
- Q Fluid flowrate
- H Height
- G Flowrate of gas
- L Flowrate of liquid

Abstract

Gas turbine power plants are widely used for power generation in the world; they are low cost, quick to install and engender stability with regard to electricity grid variations. They are nevertheless negatively impacted by ambient temperature: on hot days power demand increases while gas turbine power falls. An 18% decrease in efficiency occurs at ambient temperature 40◦C due to lower air density and the resulting increase in compressor specific work. Inlet cooling methods are used to cool inlet air to boost the power loss on hot days.

Our project presents different options for gas turbine inlet air cooling which includes evaporative air cooling, absorption chiller etc. The cooling method proposed in the project provides maximum power output as compared to its counterparts. The method of defining power gain caused by air cooling, as well as the results of applying air cooling to several different gas turbines and one gas turbine in combined cycle in domestic ambient conditions are presented and discussed.

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Chapter 1: Introduction

1.1 Pak Arab History:

Pak Arab is a subsidy of a Fatima Group which is one of the largest conglomerate groups of Pakistan. For the last 38 years, Pak Arab Fertilizers Limited has been the only fertilizer company in Pakistan producing compound fertilizers, Calcium Ammonium Nitrate (CAN) and Nitro Phosphate (NP). The Plant also produces Urea.

Fatima Energy plans to play a substantial role in overcoming the energy crisis confronting the economy by installing a 120 MW power generation plant.

Under the new management, Pak Arab Fertilizers Limited has undergone extensive modernization and new improved processes have been introduced to maximize the output while minimizing the negative impacts on the environment. For this a Clean Development Mechanism (CDM) plant was installed, which is the first project of this kind in Pakistan. Basic aim of this project is the abatement of N_2O and NOX emissions from the stack gases of Nitric Acid plant. The reduction of greenhouse effect of these gases shows the new management's commitment towards a cleaner environment.

1.2 Introduction to Gas Turbine:

A gas turbine is a combustion engine that can convert natural gas or other liquid fuels to mechanical energy. This energy then drives a generator that produces electrical energy. It is electrical energy that moves along power lines to homes and businesses.

1.3 Operating Principle of Gas turbine:

A gas turbine works in the following way:

• it draws in air from the surrounding environment;

• it compresses it to a higher pressure;

• it increases the energy level of the compressed air by adding and burning fuel in a combustion chamber;

• it directs high pressure, high temperature air to the turbine section, which converts thermal energy into mechanical energy that makes the shaft revolve; this serves, on the one hand, to supply useful energy to the driven machine, coupled to the machine by means of a coupling and, on the other hand, to supply energy necessary for air compression, which takes place in a compressor connected directly with the turbine section;

• it exhausts low pressure, low temperature gases resulting from the abovementioned transformation into the atmosphere.

1.4 Operating Cycle:

The basic operation of the gas turbine is a Brayton cycle with air as the working fluid. Fresh atmospheric air flows through the compressor that brings it to higher pressure. Energy is then added by spraying fuel into the air and igniting it so the combustion generates a high-temperature flow. This high-temperature high-pressure gas enters a turbine, where it expands down to the exhaust pressure, producing a shaft work output in the process. The turbine shaft work is used to drive the compressor; the energy that is not used for shaft work comes out in the exhaust gases that produce thrust. The purpose of the gas turbine determines the design so that the most desirable split of energy between the thrust and the shaft work is achieved. The fourth step of the Brayton cycle (cooling of the working fluid) is omitted, as gas turbines are open systems that do not use the same air again.

Gas turbines are used to power aircraft, trains, ships, electrical generators, pumps, gas compressors and tanks.

1.5 Brayton Cycle:

The thermodynamic cycle of a gas turbine is known as the Brayton cycle. Fig. Th.4 illustrates a diagram of a gas turbine (in this specific case, an double shaft turbine). This diagram is useful to understand the meaning of the thermodynamic cycle more easily.

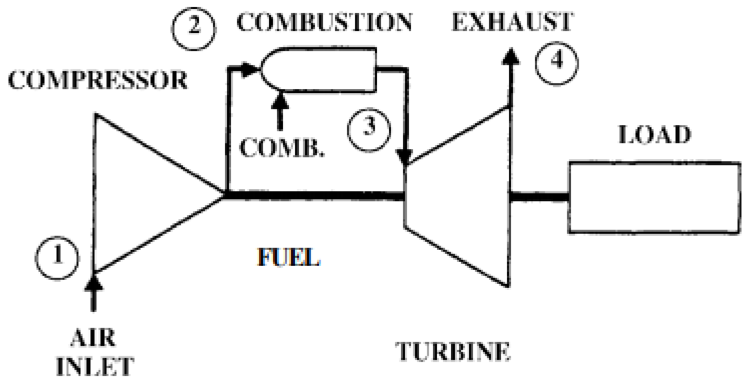


Figure 1: Gas Turbine Operating Cycle

Air enters the compressor at point (1), which represents atmospheric air conditions. These conditions are classified according to pressure, temperature and relative humidity values. Standard design conditions are conventionally classified as ISO Conditions, with the following reference values:

ISO CONDITIONS

Ambient temperature (°C) 15

Ambient pressure (mbar) 1013

Relative humidity (%) 60

The air is compressed inside the compressor and exits in the condition indicated at point (2). During the transformation from (1) to (2), no heat is transferred to the air but. Air temperature increases, due to polytropic compression, up to a value that varies depending on gas turbine model and ambient temperature.

After leaving the compressor, air enters the combustion area, practically under the same pressure and temperature conditions as at point (2) (except for losses undergone on the way from the compressor discharge to the combustion chamber inlet, which amount to about 3 to 4% of the absolute value of discharge pressure).

Fuel is injected into the combustion chamber via a burner, and combustion takes place at practically constant pressure. The transformation between points (2) and (3) represents not only combustion. In fact, the temperature of the actual combustion process, which takes place under virtually stochiometric conditions, reaches values (around 2000°C) locally, in the combustion area next to the burner, which are too high for the resistance of materials downstream. Therefore, the final temperature of the transformation relative to point (3), is lower, as it is the result of mixing the primary combustion gases with cooling and dilution air as described previously. In this regard, it is useful to give some definitions of temperature at point (3), which is the maximum cycle temperature or firing temperature.

The following transformation, comprised between points (3) and (4), represents the expansion of gases through the turbine section, which, as mentioned before, converts thermal energy and pressure into kinetic energy and, by means of shaft rotation, into work used for compression (internal, not usable) and external useful work, through coupling with a driven machine. Over 50% of the energy developed by expansion in the gas turbine is required for the compression by the axial compressor. Downstream of section (4), gases are exhausted into the atmosphere. The thermodynamic representation of the events described so far is visible in Fig. Th.6 (pressure diagrams - volume P-V and temperature - entropy T-S).

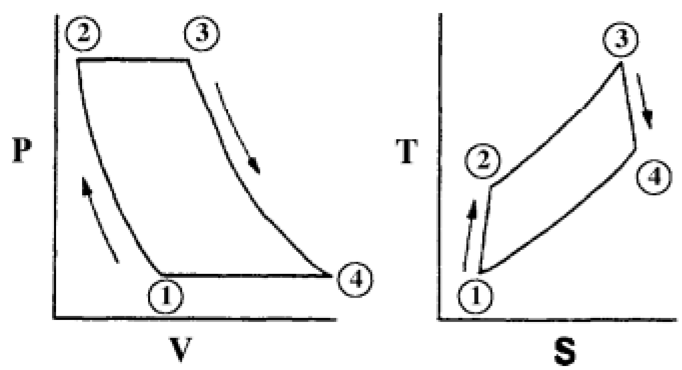


Figure 2: The Brayton Cycle

In the cycle illustrated in the above figure, the 4 points correspond to the same points described before.

1.6 ENGINE SECTIONS

1.6.1 Inlet

The air inlet duct must provide clean and unrestricted airflow to the engine. Clean and undisturbed inlet airflow extends engine life by preventing erosion, corrosion, and foreign object damage (FOD).

Consideration of atmospheric conditions such as dust, salt, industrial pollution, foreign objects (birds, nuts and bolts), and temperature (icing conditions) must be made when designing the inlet system. Fairings should be installed between the engine air inlet housing and the inlet duct to ensure minimum airflow losses to the engine at all airflow conditions.

The inlet duct assembly is usually designed and produced as a separate system rather than as part of the design and production of the engine.

1.6.2 Compressor

The compressor is responsible for providing the turbine with all the air it needs in an efficient manner. In addition, it must supply this air at high static pressures. The example of a large turboprop axial flow compressor will be used. The compressor is Engine Temperature and Pressure Flow 6 assumed to contain fourteen stages of rotor blades and stator vanes. The overall pressure ratio (pressure at the back of the compressor compared to pressure at the front of the compressor) is approximately 9.5:1. At 100% (>13,000) RPM, the engine compresses approximately 433 cubic feet of air per second. At standard day air conditions, this equals approximately 33 pounds of air per second. The compressor also raises the temperature of the air by about 550°F as the air is compressed and moved rearward. The power required to drive a compressor of this size at maximum rated power is approximately 7000 horsepower.

In an axial flow compressor, each stage incrementally boosts the pressure from the previous stage. A single stage of compression consists of a set of rotor blades attached to a rotating disk, followed by stator vanes attached to a stationary ring. The flow area between the compressor blades is slightly divergent. Flow area between compressor vanes is also divergent, but more so than for the blades.

In general terms, the compressor rotor blades convert mechanical energy into gaseous energy. This energy conversion greatly increases total pressure (Pt). Most of the increase is in the form of velocity (Pi), with a small increase in static pressure (Ps) due to the divergence of the blade flow paths.

The stator vanes slow the air by means of their divergent duct shape, converting 'the accelerated velocity (Pi) to higher static pressure (Ps). The vanes are positioned at an angle such that the exiting air is directed into the rotor blades of the next stage at the most efficient angle. This process is repeated fourteen times as the air flows from the first stage through the fourteenth stage. Figure 2-4 shows one stage of the compressor and a graph of the pressure characteristics as the air flows through the stage.

In addition to the fourteen stages of blades and vanes, the compressor also incorporates the inlet guide vanes and the outlet guide vanes. These vanes, located at the inlet and the outlet of the compressor, are neither divergent nor convergent. The inlet guide vanes direct air to the first stage compressor blades at the "best" angle. The outlet guide vanes "straighten" the air to provide the combustor with the proper airflow direction.

The efficiency of a compressor is primarily determined by the smoothness of the airflow. During design, every effort is made to keep the air flowing smoothly through the compressor to minimize airflow losses due to friction and turbulence. This task is a difficult one, since the air is forced to flow into ever-higher pressure zones.

Air has the natural tendency to flow toward low-pressure zones. If air were allowed to flow "backward" into the lower pressure zones, the efficiency of the compressor would decrease tremendously as the energy used to increase the pressure of the air was wasted. To prevent this from occurring, seals are incorporated at the base of each row of vanes to prevent air leakage. In addition, the tip clearances of the rotating blades are also kept at a minimum by the use of coating on the inner surface of the compressor case. All components used in the flow path of the compressor are shaped in the form of airfoils to maintain the smoothest airflow possible. Just as is the case for the wings of an airplane, the angle at which the air flows across the airfoils is critical to performance.

The blades and vanes of the compressor are positioned at the optimum angles to achieve the most efficient airflow at the compressor’s maximum rated speed. Any deviation from the maximum rated speed changes the characteristics of the airflow within the compressor. The blades and vanes are no longer positioned at their optimum angles. Many engines use bleed valves to unload the force of excess air in the compressor when it operates at less than optimum speed. The example engine incorporates four bleed valves at each of the fifth and tenth compressor stages. They are open until 13,000 RPM (~94% maximum) is reached and allow some of the compressed air to flow out to the atmosphere. This results in higher air velocities over the blade and vane airfoils, improving the airfoil angles. The potential for airfoil stalling is reduced, and compressor acceleration can be accomplished without surge.

1.6.3 Diffuser

Air leaves the compressor through exit guide vanes, which convert the radial component of the air flow out of the compressor to straight-line flow. The air then enters the diffuser section of the engine, which is a very divergent duct. The primary function of the diffuser structure is aerodynamic. The divergent duct shape converts most of the air's velocity (Pi) into static pressure (PS). As a result, the highest static pressure and lowest velocity in the entire engine is at the point of diffuser discharge and combustor inlet. Other aerodynamic design considerations that are important in the diffuser section arise from the need for a short flow path, uniform flow distribution, and low drag loss. In addition to critical aerodynamic functions, the diffuser also provides:

- Engine structural support, including engine mounting to the nacelle
- Support for the rear compressor bearings and seals 8
- Bleed air ports, which provide pressurized air for:
- airframe "customer" requirements (air conditioning, etc.)
- engine inlet anti-icing
- control of acceleration bleed air valves
- Pressure and scavenge oil passages for the rear compressor and front turbine bearings.
- Mounting for the fuel nozzles.

1.6.4 Combustor

Once the air flows through the diffuser, it enters the combustion section, also called the combustor. The combustion section has the difficult task of controlling the burning of large amounts of fuel and air. It must release the heat in a manner that the air is expanded and accelerated to give a smooth and stable stream of uniformly-heated gas at all starting and operating conditions. This task must be accomplished with minimum pressure loss and maximum heat release. In addition, the combustion liners must position and control the fire to prevent flame contact with any metal parts. The engine in this example uses a can-annular combustion section. Six combustion liners (cans) are positioned within an annulus created by inner and outer combustion cases. Combustion takes place in the forward end or primary zone of the cans. Primary air (amounting to about one fourth of the total engine's total airflow) is used to support the combustion process. The remaining air, referred to as secondary or dilution air, is admitted into the liners in a controlled manner. The secondary air controls the flame pattern, cools the liner walls, dilutes the temperature of the core gasses, and provides mass. This cooling air is critical, as the flame temperature is above 1930°C (3500'F), which is higher than the metals in the engine can endure. It is important that the fuel nozzles and combustion liners control the burning and mixing of fuel and air under all conditions to avoid excess temperatures reaching the turbine or combustion cases. Maximum combustion section outlet temperature (turbine inlet temperature) in this engine is about 1070°C (>1950°F). The rear third of the combustion liners is the transition section. The transition section has a very convergent duct shape, which begins accelerating the gas stream and reducing the static pressure in preparation for entrance to the turbine section.

1.6.5 Turbine

This example engine has a four-stage turbine. The turbine converts the gaseous energy of the air/burned fuel mixture out of the combustor into mechanical energy to drive the compressor, driven accessories, and, through a reduction gear, the propeller. The turbine converts gaseous energy into mechanical energy by expanding the hot, high-pressure gases to a lower temperature and pressure. Each stage of the turbine consists of a row of stationary vanes followed by a row of rotating blades. This is the reverse of the order in the compressor.

In the compressor, 9 energies is added to the gas by the rotor blades, then converted to static pressure by the stator vanes. In the turbine, the stator vanes increase gas velocity, and then the rotor blades extract energy. The vanes and blades are airfoils that provide for a smooth flow of the gases. As the airstream enters the turbine section from the combustion section, it is accelerated through the first stage stator vanes. The stator vanes (also called nozzles) form convergent ducts that convert the gaseous heat and pressure energy into higher velocity gas flow (Pi). In addition to accelerating the gas, the vanes "turn" the flow to direct it into the rotor blades at the optimum angle. As the mass of the high velocity gas flows across the turbine blades, the gaseous energy is converted to mechanical energy. Velocity, temperature, and pressure of the gas are sacrificed in order to rotate the turbine to generate shaft power. Figure 2-5 represents one stage of the turbine and the characteristics of the gases as it flows through the stage.

The efficiency of the turbine is determined by how well it extracts mechanical energy from the hot, high-velocity gasses. Since air flows from a high-pressure zone to a low pressure zone, this task is accomplished fairly easily. The use of properly positioned airfoils allows a smooth flow and expansion of gases through the blades and vanes of the turbine. All the air must flow across the airfoils to achieve maximum efficiency in the turbine. In order to ensure this, seals are used at the base of the vanes to minimize gas flow around the vanes instead of through the intended gas path.

In addition, the first three stages of the turbine blades have tip shrouds to minimize gas flow around the blade tips. Exhaust After the gas has passed through the turbine, it is discharged through the exhaust. Though most of the gaseous energy is converted to mechanical energy by the turbine, a significant amount of power remains in the exhaust gas. This gas energy is accelerated through the convergent duct shape of the exhaust to make it more useful 10 as jet thrust - the principle of equal and opposite reaction means that the force of the exhausted air drives the airplane forward.

1.7 Problem Statement

As the compressor inlet temperature increases, the specific compression work increases, while the weight flow rate of the air decreases (because of a decrease in specific weight g). Consequently, the turbine efficiency and useful work (and, therefore, power) decrease. If temperature decreases, the reverse occurs. Our project is to devise a method to cool the inlet air coming into the compressor in order to increase the turbine efficiency. The project entails studies of the effects of ambient air temperature on gas turbine efficiency in open cycles as well as combined cycles proposed to increase the overall turbine efficiency. The methods that were taken into consideration were:

- Evaporative Cooling Method
- Mechanical Chilling Method

1.8 Objectives

This research and evaluation study will focus on the possible performance enhancement of the reference gas turbine by supplying charge air cooling. Main objective is to derive the behavior of the gas turbine with the ambient temperature variation. A thermodynamic model for the inlet air cooling performance will then be developed and approximated by simulations. Further, the use of available exhaust gas heat to drive the charge air cooler and its technical and economic feasibility will be studied for analyzing the adoption of a practical absorption refrigeration system.

Chapter 2 Literature Review

2.1 EFFECTS OF ATMOSPHERIC CONDITIONS

The performance of the gas turbine engine is dependent on the mass of air entering the engine. At a constant speed, the compressor pumps a constant volume of air into the engine with no regard for air mass or density. If the density of the air decreases, the same volume of air will contain less mass, so less power is produced. If air density increases, power output also increases as the air mass flow increases for the same volume of air. Atmospheric conditions affect the performance of the engine since the density of the air will be different under different conditions. On a cold day, the air density is high, so the mass of the air entering the compressor is increased. As a result, higher horsepower is produced. In contrast, on a hot day, or at high altitude, air density is decreased, resulting in a decrease of output shaft power.

Gas turbines working in open cycle are engines which are sensitive to ambient temperature changes. The increase in inlet air temperature, especially pronounced in summer, causes a significant decrease in gas turbine output power. The reduction of inlet air temperature can be achieved by the application of air cooling through water atomization or installing a chiller in the inlet ducting. The use of a cooler or chiller is economically justified if the profits from the increase in power exceed the related capital and operating costs and the climate conditions for effective operation is met. Evaporative cooling, achieved by evaporation of water injected into the inlet duct, is a cost-effective way of recovering turbine capacity during high temperature and low or moderate humidity periods. This method is hindered by ambient temperatures of 10–15◦C and above. In lower temperatures there is a higher risk of ice formation on the compressor's inlet guide vanes. The effectiveness of cooling is determined by the ratio of the reduction in the temperature in the cooler to the largest possible reduction, when the wet bulb temperature is achieved. The actual drop in temperature depends on the device's construction and the atmospheric conditions.

A gas turbine uses atmospheric air, therefore, its performance is greatly affected by all factors that influence the weight flow rate of air delivered to the compressor. These factors are:

• Temperature

• Pressure

• Relative humidity

In this regard, we remind you that reference conditions for the three above-mentioned factors are, by convention, ISO standards. As the compressor inlet temperature increases, the specific compression work increases, while the weight flow rate of the air decreases (because of a decrease in specific weight g). Consequently, the turbine efficiency and useful work (and, therefore, power) decrease. If temperature decreases, the reverse occurs. This tie between compressor inlet temperature and power and efficiency varies from turbine to turbine, according to cycle parameters, compression and expansion efficiencies and air flow rate. Fig.3 shows an example of how power, heat rate and exhaust gas flow are affected by ambient temperature.

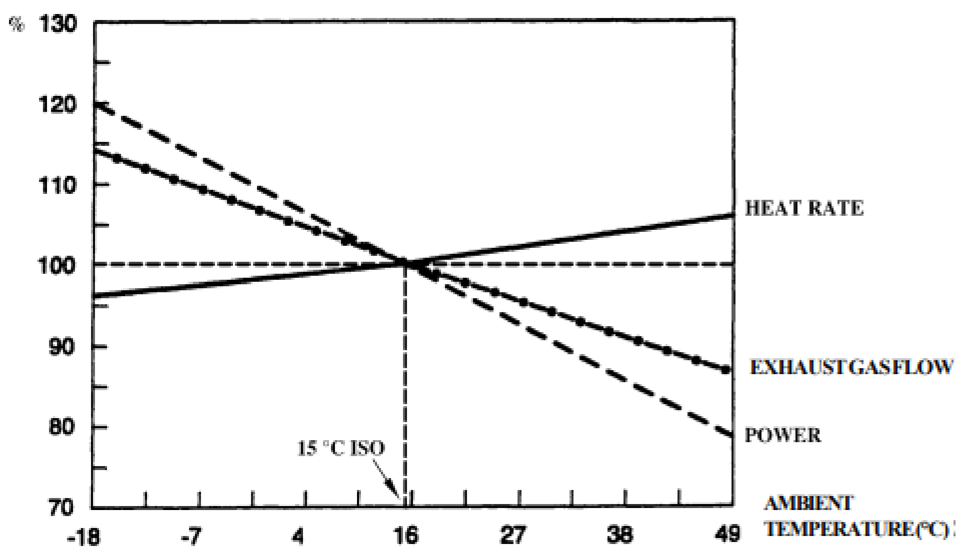


Figure 3: Influence of ambient temperature on gas turbine performance

2.2 Evaporative Cooling

The curves in Fig 3 clearly show how power and efficiency increase as the compressor air inlet temperature decreases.

The latter can be reduced artificially by using an evaporative cooler located upstream of the suction filter.

Water separated into drops or in the form of a liquid film, cools the air by evaporating in the cooler as it flows in the opposite direction, thus originating an adiabatic iso-enthalpic exchange.

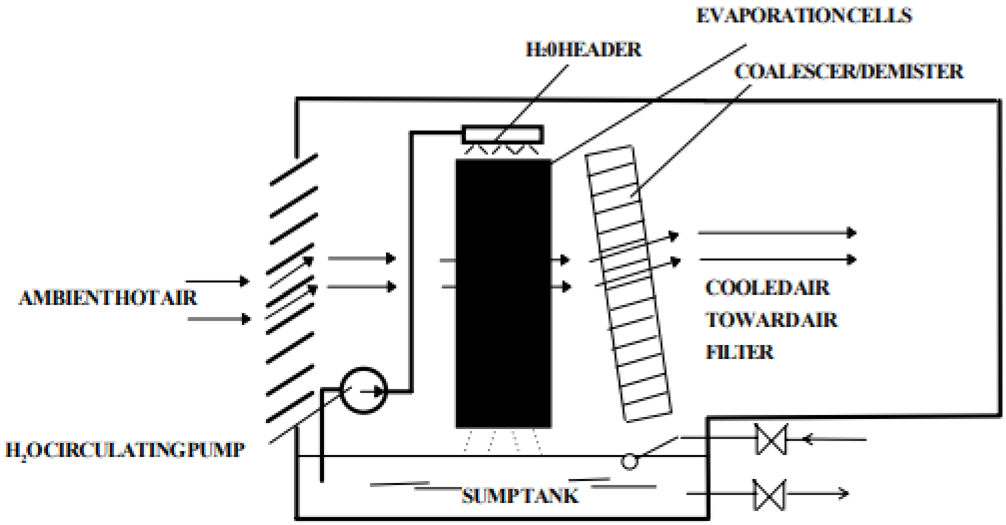


Figure 4: Evaporative Cooler

In order to prevent water from drawing towards compressor and fouling it, downstream of the cooler there are one or more stages of drop separators (demisters), which separate by inertia any water drops that might be entrained downstream of the cooler by the flow of air ingested by the turbine.

2.3 Inlet Chilling

In environments with a high degree of average relative humidity (higher than 60%) and without extreme ambient temperatures, it is advisable to cool air with a different method, commonly called "inlet chilling"; according to this method, air is cooled during a refrigerating cycle (based generally on absorption) carried out in a closed loop arrangement. In this way, the restrictions imposed by relative humidity and by ambient temperature, described in the preceding system, can be eliminated. The minimum temperature reached by air at the end of the cooling process is strictly dependent on the capability of the refrigerating cycle to produce cold liquid and on the efficiency of the thermal exchange that takes place in the water - air exchanger.
Figure 5 shows an operating diagram of this system (in this example, steam is used for the absorption cycle), composed of a chiller, water connecting piping and a water - air exchanger,

installed downstream of the gas turbine inlet filter. As in evaporative cooling, also in this case it is necessary to install a coalesce/demister downstream of the system, in order to prevent moisture from reaching the compressor inlet section.

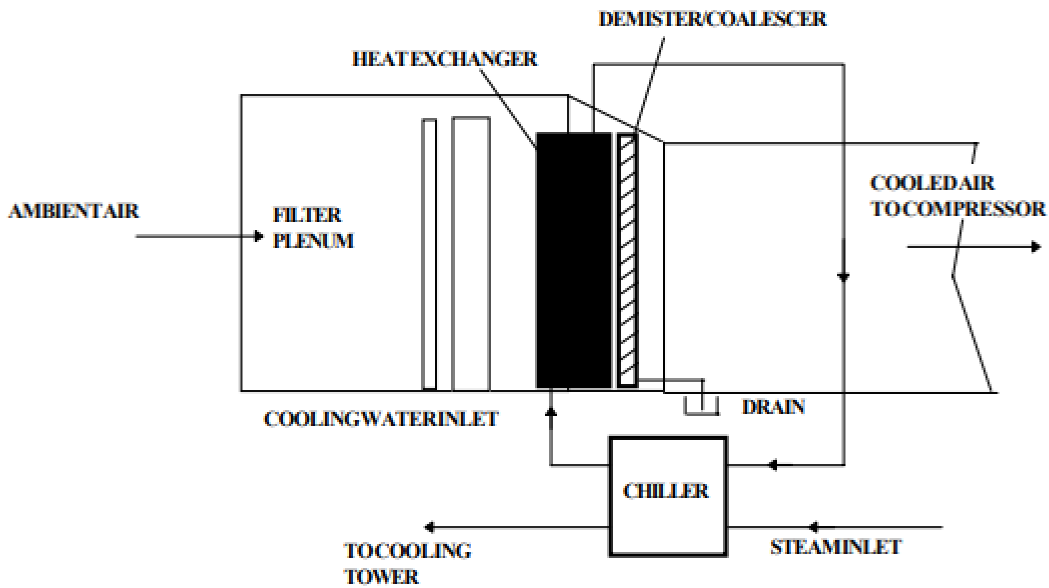


Figure 5: Air chilling/cooling system based on absorption

Figure 6 shows a comparison between the cooling powers of the two systems.

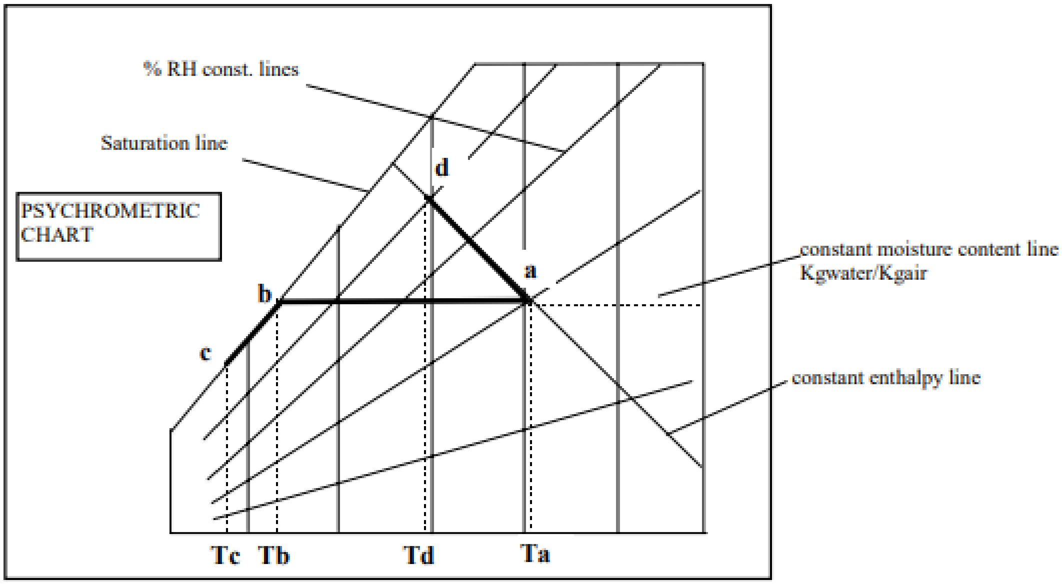


Figure 6: Comparison Psychometric Chart

Line a - d represents air cooling in the case of evaporative cooling. As mentioned before, this line follows the constant enthalpy line, resulting in a progressive increase in relative humidity.

The restriction imposed by this cooling method consists in the fact that there remains a minimum distance from the saturation curve, compatibly with realistic exchange surfaces, considered from the point of view of construction. Normal values indicate around 90% relative humidity, that is, there still remains a 10% margin before the saturation line is reached.

Under these conditions, the final air temperature is equal to Td. In the case of the chilling process, the cooling line has a constant moisture content along segment a - b. If the potential of the refrigerating cycle and the efficiency of the exchanger allow it, cooling can reach the saturation line and follow it along segment b - c, in which heat is removed to form condensate (H2 O). In this second segment, there is a smaller temperature reduction, because most of the cooling energy serves for the condensing process and only a small part of it participates in lowering temperature.

In the chilling system, the final air temperature will be equal to Tb or Tc, depending on to the selected degree of cooling.

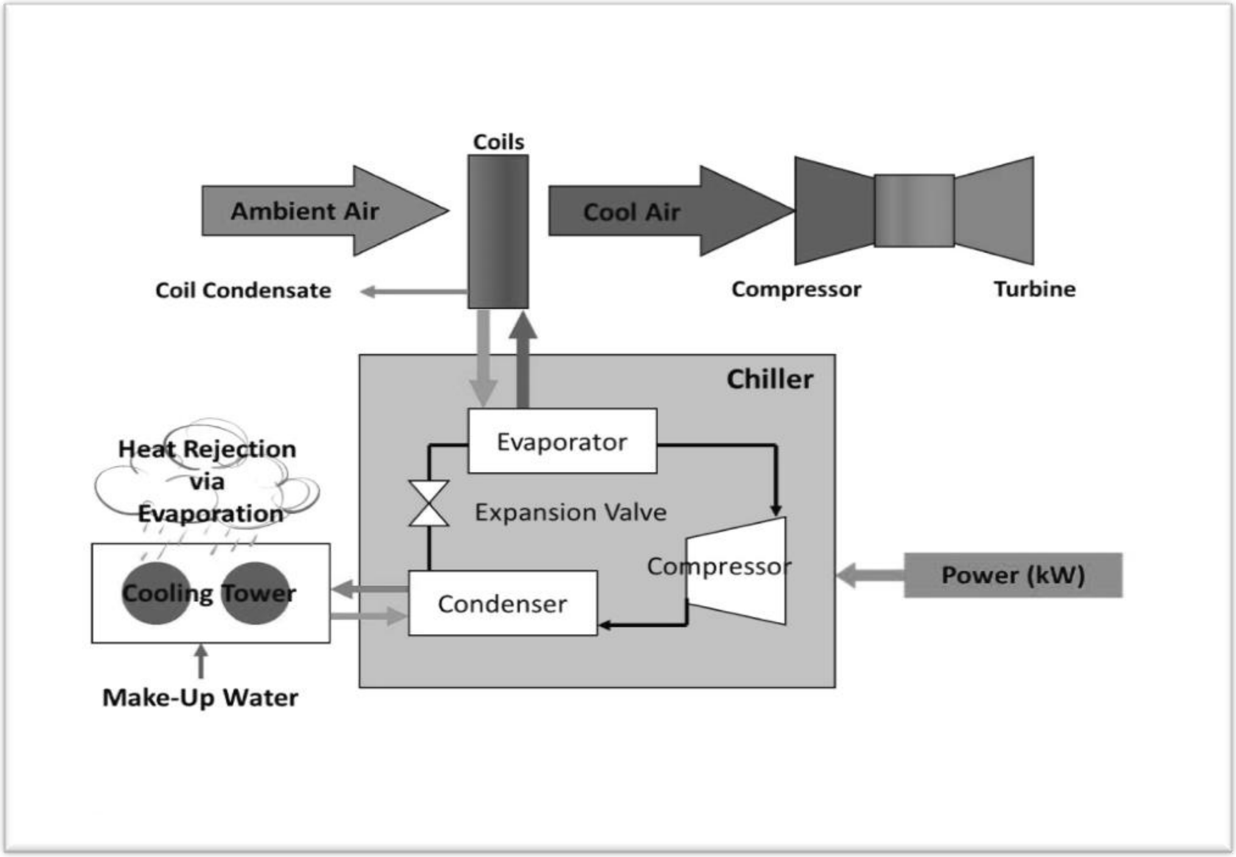


Figure 7 Inlet air cooling using mechanical chilling

Fig 7 shows graphically the variation of power output and efficiency respectively with ambient air temperature for the modeled generic gas turbine cycle.

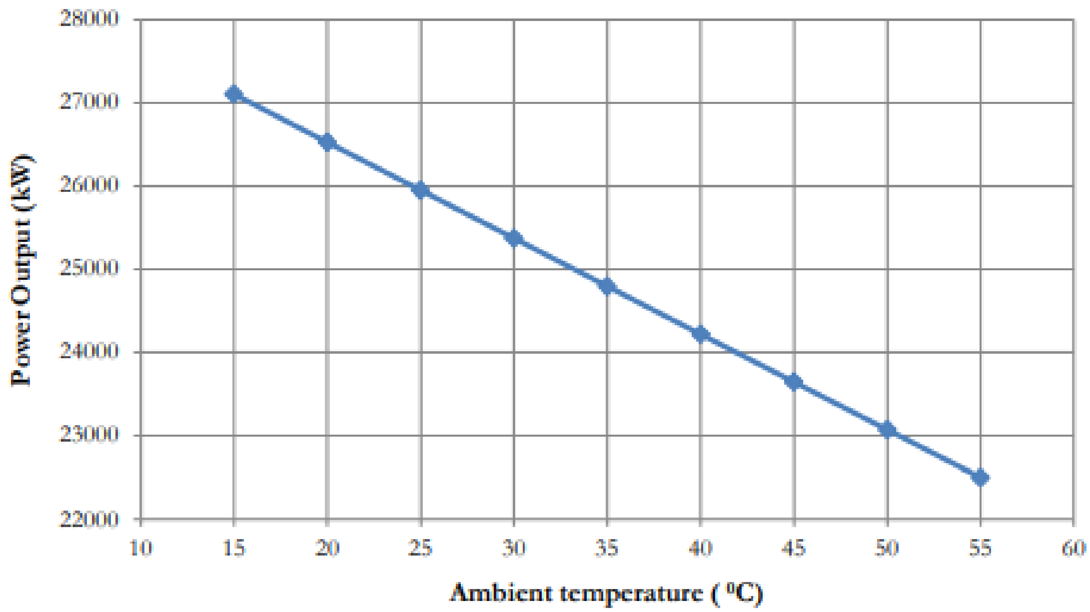


Figure 8: Power Output variation with ambient air temperature

The graph of power output is almost linear with inlet temperature, but in fact it is curved with a slight slope which is decreasing towards the increasing temperature. Average reduction in power output per degree Celsius is about 115 kW. Cooling of inlet air from 27C down to 15C would increase the power output by 1.38 MW.

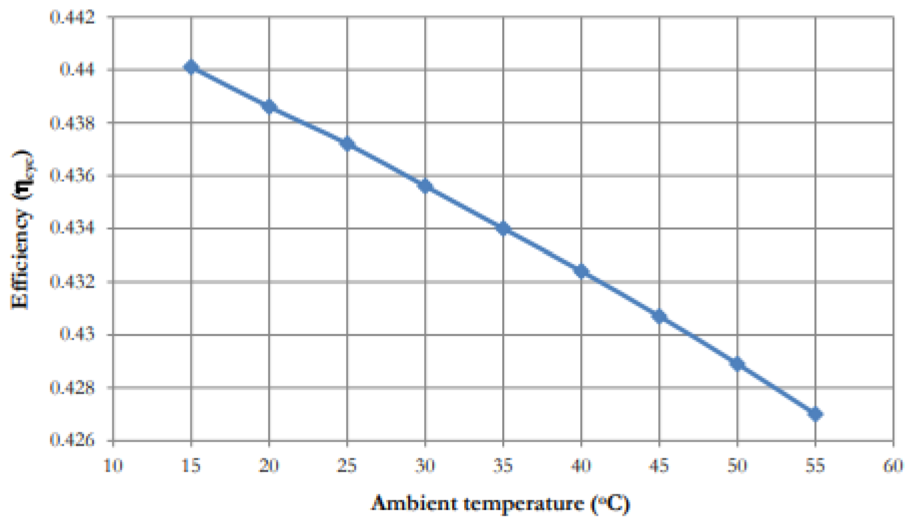


Figure 9: Efficiency variation with ambient air temperature

Chapter 3 HYSYS SIMULATION

Introduction

Power generation area of Pak Arab Fertilizers is currently working on a single loop Brayton Cycle. Gas turbine system in a single cycle generally runs with an efficiency of 35%-45%. This efficiency range varies on the basis of Inlet air temperature, Compression ratio, Rate of fuel injection, Air to fuel ratio, heat and mechanical work losses, Temperature profile in the combustor, and Mechanical design of turbine. Keeping these factors and current operating conditions, power generation area was simulated to get mathematical model. This model was further used to compare results with the modified model.

Sections of Gas turbine system:

Power generation area can be divided into following sections for simulation:

- Compression section
- Combustion section
- Air Division section
- Expansion section
- Exhaust

Operation Conditions and Mechanism:

Air fed to the compressor is compressed at a ratio of 12:1. This compression is achieved in 8 stages in real time gas turbine system but our simulated model assumes 2 stages. This compressed air divides into two air streams, Diluent and combustion air. 65% of the total air is directed towards the combustion chamber where heat energy is added by fuel combustion ultimately increasing air temperature and pressure head. This energized air feed is sent into expansion section where work is produced using 8 staged turbine system. Exhaust with unrecovered thermal energy is sent into environment.

Flaws in Existing model:

Existing single cycle system is less efficient due to thermal losses in exhaust and high compression load due to high feed temperature. Variation in ambient air temperature affects the exhaust temperature of combustion chamber. Oxygen density entering the system is reduced hence adding less heat to the feed stream. Unrecovered energy in the exhaust results in global warming and also increases the operation cost of the cycle. Existing working model of Power generation area is show in figure 10a. This model makes the following assumptions:

1) Compression is single staged
2) Combustion chamber is a simple conversion reactor with 100% conversion rate.
3) Expansion occurs in a single stage turbine

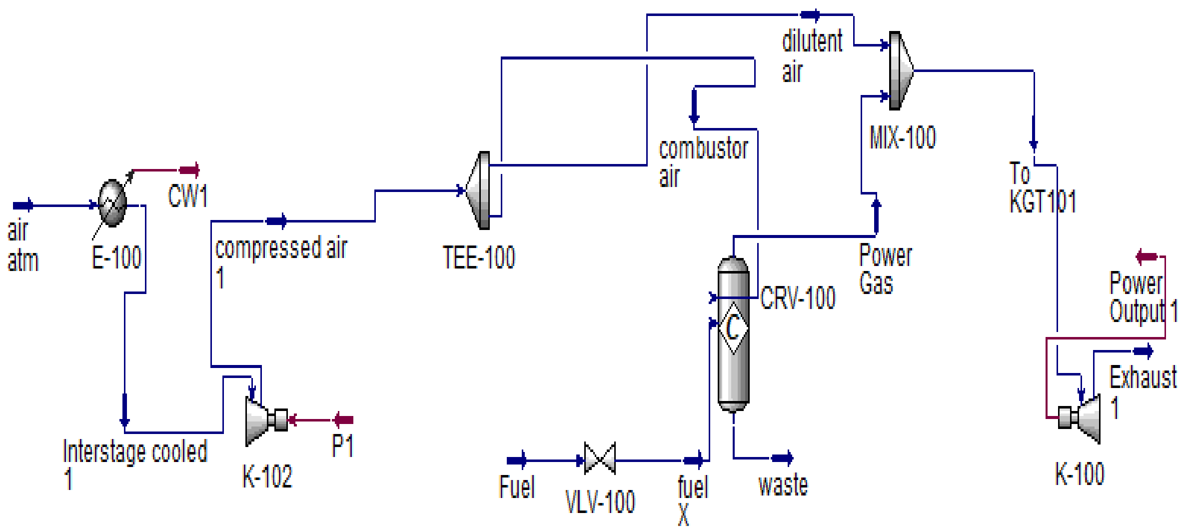


Figure 10: Process flow diagram designed of existing GT model designed on Aspen HYSYS.

Performance Parameters of the existing model:

Existing model runs with a thermal efficiency of 47.56 % and Polytropic efficiency of 71.54% giving a power output of 8980 Megawatts. Other performance parameters are shown in fig 11. Although these parameters are satisfying for a single cycle power production but this system can be made more efficient and production rate can be increased by making changes in the existing model. Inlet chilling system for this system was simulated and tested for the given conditions and successful results were achieved which are shared later in this chapter.

Results

Adiabatic Head [m]	7.244e+004
Polytropic Head [m]	7.594e+004
Adiabatic Fluid Head [kJ/kg]	710.4
Potential Fluid Head [kJ/kg]	744.7
Adiabatic Efficiency	75.000
Polytropic Efficiency	71.548
Power Produced [kW]	8980
Friction Loss [kW]	0.0000
Rotational inertia [kW]	0.0000
Fluid Power [kW]	8980
Polytropic Head Factor	1.0016
Polytropic Exponent	1.1850
Isentropic Exponent	1.2816

Figure 11: Performance parameters of the existing GT model calculated by Aspen HYSYS.

Proposed Aspen HYSYS model for Gas Turbine:

Based on the drawbacks of the existing conditions, Modification in the basic design and operating parameters were required. Following modifications are proposed in order to maximize the efficiency and power output of the system.

Modifications:

1) **Design of a Mechanical chilling system**:

Design of a mechanical chilling system for the inlet air to reduce its temperature form ambient range (32C -38C) to 15C. This is the optimum temperature is found by running a test session on the existing Simulated model. Air was fed in the system at different temperature values and their power output was calculated. Plotting these values, as shown in figure 12, gives the optimum temperature point at which discharge of chilling system is adjusted.

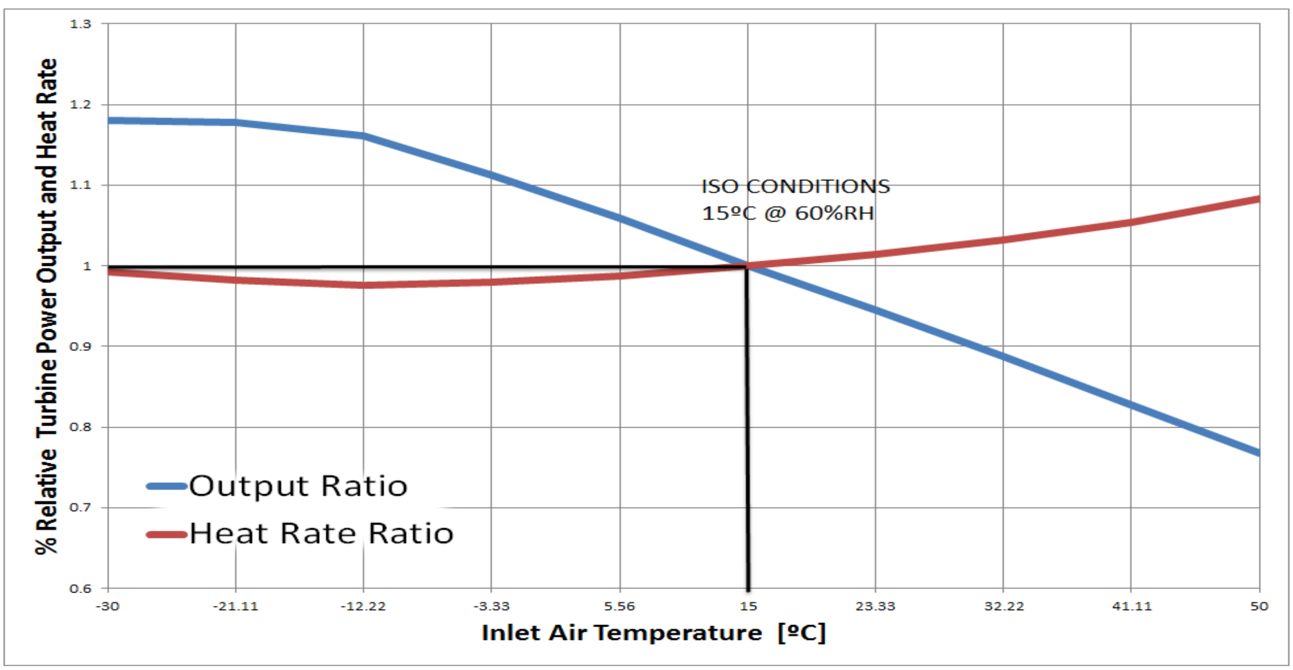


Figure 12: Plot between the power output, heat rate VS the temperature change in inlet air.

2) **Designing of heat recovery steam generation system:**

In order to recover the waste heat of exhaust stream, a series of heat exchangers were installed at the exhaust chamber. This heat is utilized to generate steam which is further superheated and used in closed Rankine cycle to produce additional 1.28 megawatts. This accommodates for the auxiliary power requirement of the chilling system and the compression load is shared.11.36 % load reduction is achieved by this model. Proposed model is shown in Fig 14 below:

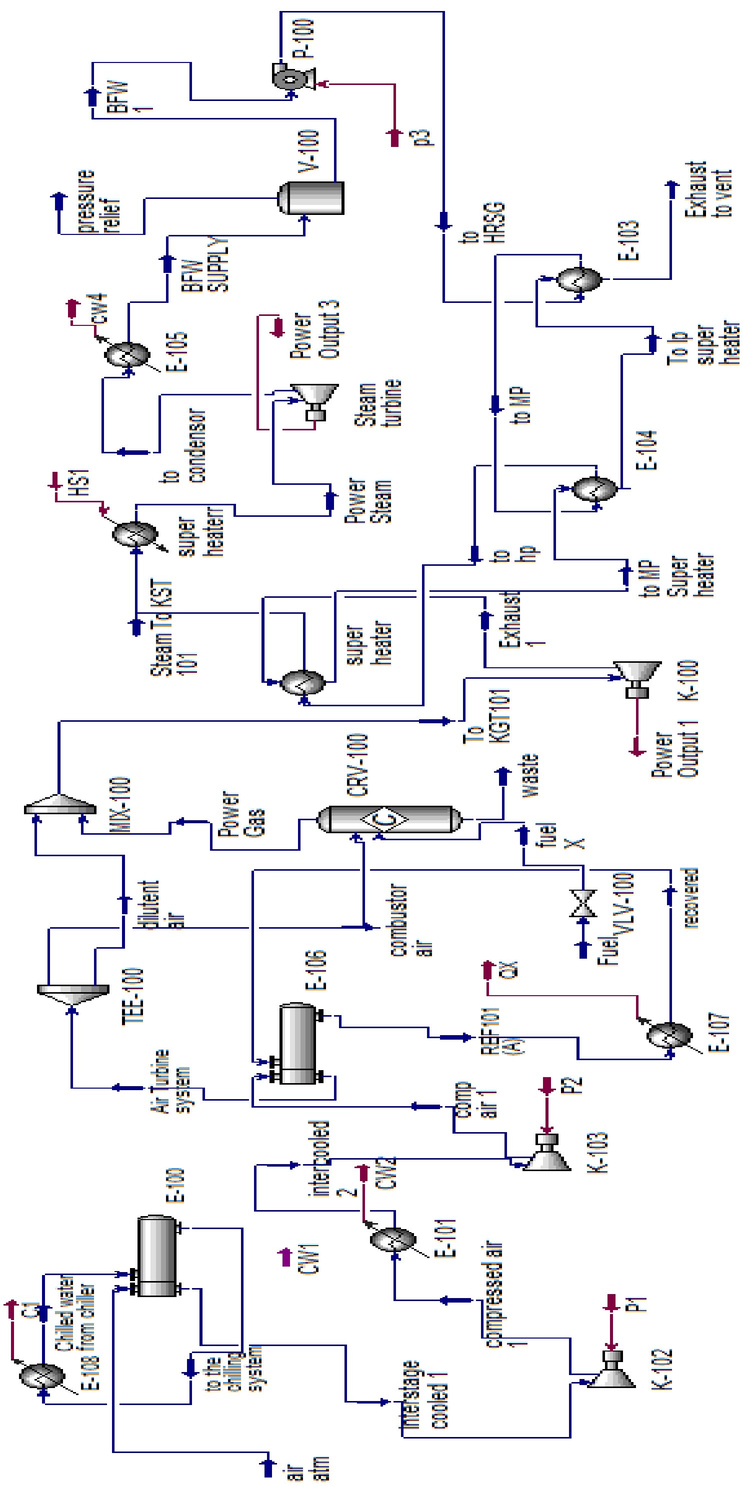


Figure 13: Proposed model with HRSG and Chilling system addition

Modeling of Gas Turbine system:

Each section of the gas turbine system is individually simulated for the given operating conditions in order to determine power requirements and efficiencies. Results obtained from modified simulated model are compared with manual energy and material balance calculation to accounts for the deviation from calculated results. Also performance parameters calculated for the proposed model as calculated by the software are the basis for comparative analysis with the existing model.

- **Compression Section:**

This section is responsible for the total power input and load on the system. Optimization can be achieved by cooling compression feed with reduces the load by 11.36%. Initial and modified performance parameters of the compression cycle are shown in figure 15 and figure 16.

Results

Adiabatic Head [m]	2.618e+004
Polytropic Head [m]	2.831e+004
Adiabatic Fluid Head [kJ/kg]	256.8
Polytropic Fluid Head [kJ/kg]	277.6
Adiabatic Efficiency	75.000
Polytropic Efficiency	81.102
Power Consumed [kW]	5459
Polytropic Head Factor	1.0021
Polytropic Exponent	1.5236
Isentropic Exponent	1.3916
Speed [rpm]	<empty>

Figure 14a: Modified Performance Parameters Of compressor.

Results

Adiabatic Head [m]	2.816e+004
Polytropic Head [m]	3.044e+004
Adiabatic Fluid Head [kJ/kg]	276.2
Polytropic Fluid Head [kJ/kg]	298.5
Adiabatic Efficiency	75.000
Polytropic Efficiency	81.062
Power Consumed [kW]	5873
Polytropic Head Factor	1.0023
Polytropic Exponent	1.5186
Isentropic Exponent	1.3881
Speed [rpm]	<empty>

Figure 15: Existing Compression Parameters

- **Mechanical chilling system:**

This system consists of evaporating and condensing end. Fig 13 shows the stream connections into the mechanical chilling system. Compressor feed passes through a tube bundle installed in the air duct to decrease its temperature. This feed rejects its heat into water being circulated in the tube bundle. Water stream with added heat from the feed goes to the condenser where Refrigerant R134A absorbs the heat and evaporates. This refrigerant is further passed through its refrigeration cycle so that added heat can be rejected to the environment/cooling water. Operating conditions of the chiller are shown in figure 16. Shell and tube orientations are shown in figure 17.

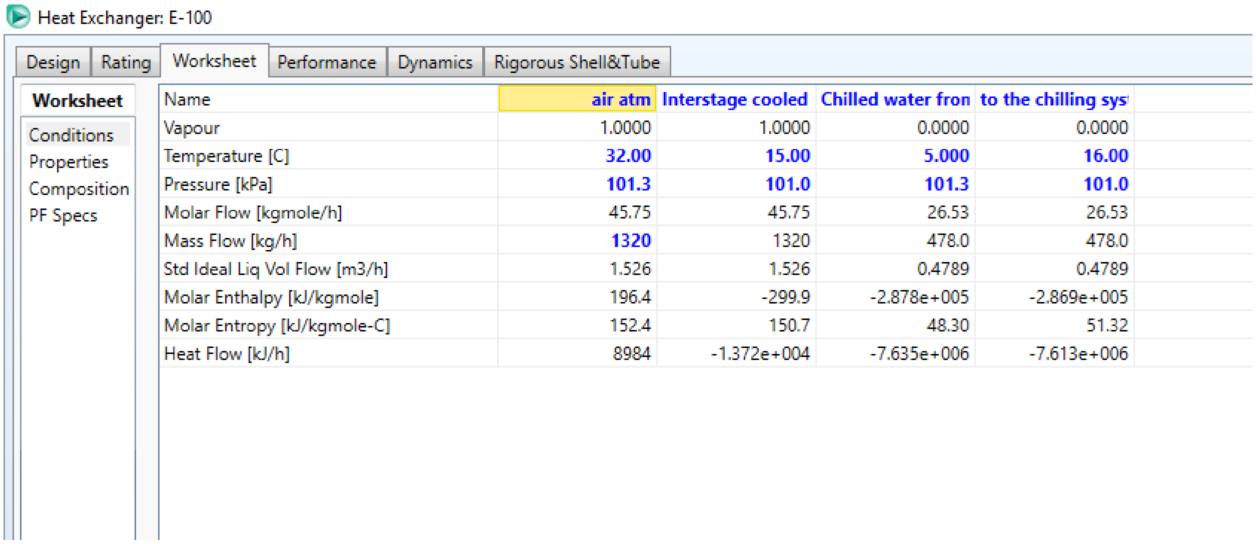


Figure 16: Operating Conditions for chiller.

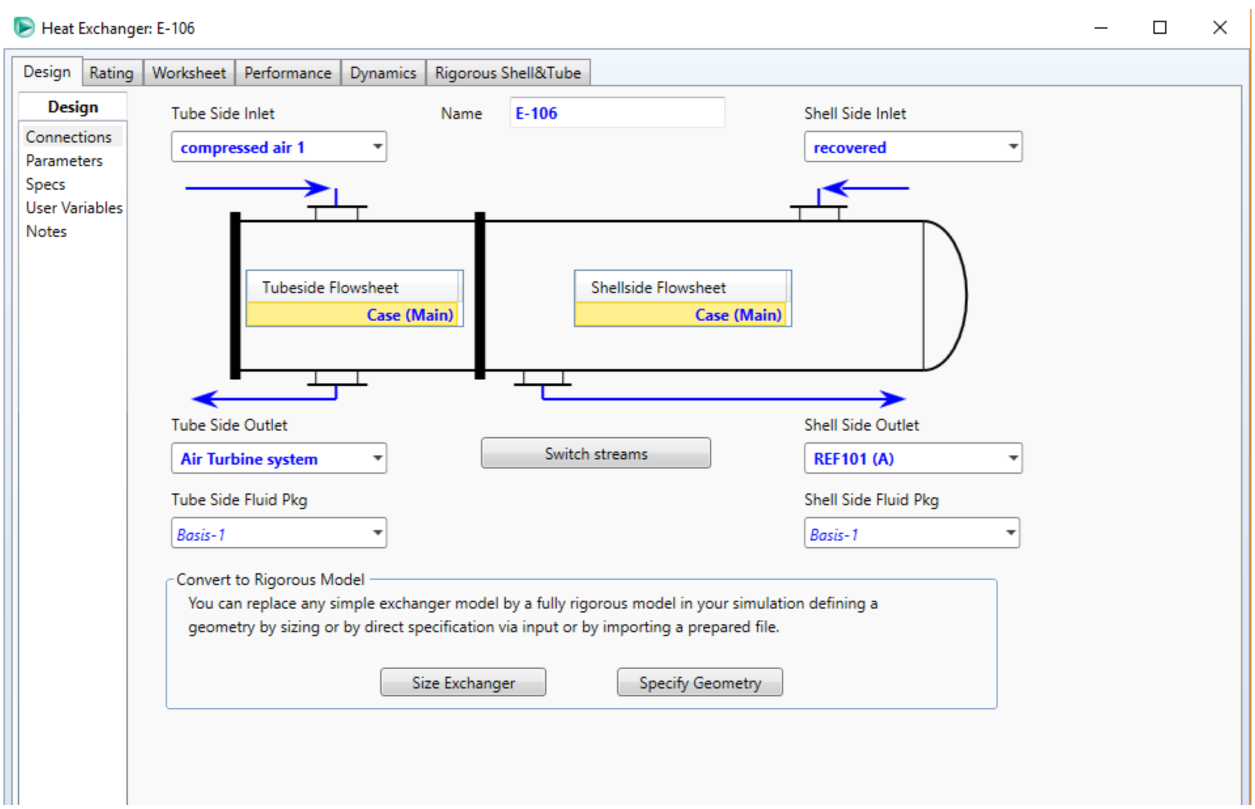


Figure 17 Shell and Tube stream orientation for chiller.

- **Combustion Section:**

In this section, Fuel injection is done which results into combustion reaction. This reaction adds heat to the air stream and hence temperature of air is significantly increased as shown in the figure 19. Overall reaction kinetics , as shown in figure 18, are achieved by modeling this section. These parameters help in concluding the final efficiencies of the system.

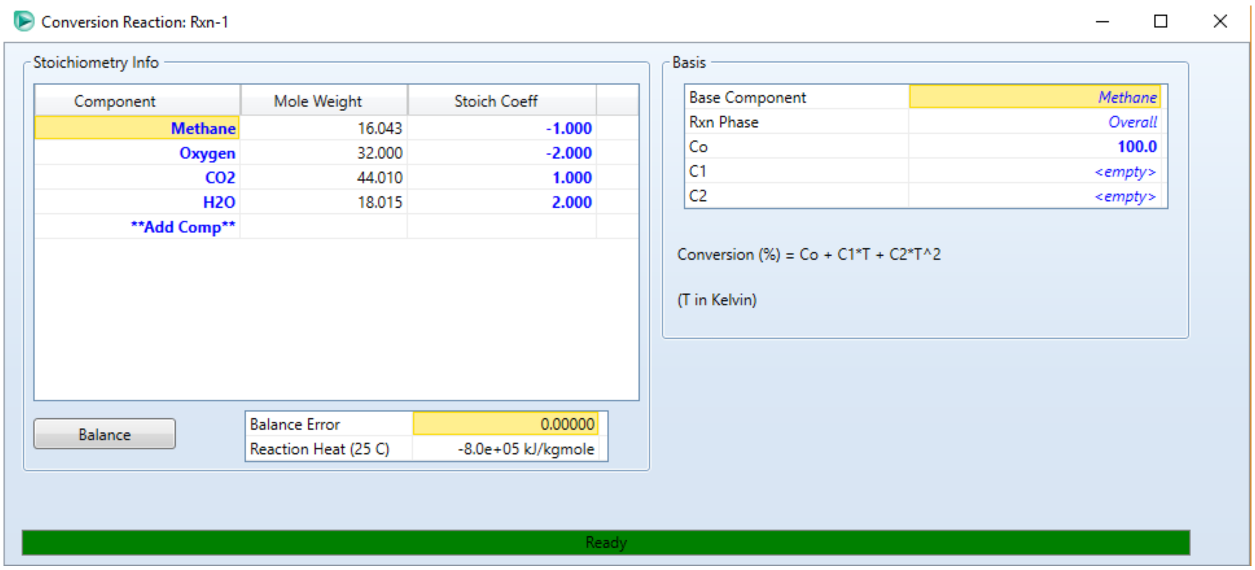


Figure 18: Stiochiometry Of combustion reaction

Name	combustor air	fuel X	waste	Power Gas
Vapour	1.0000	1.0000	0.0000	1.0000
Temperature [C]	169.0	31.39	1887	1887
Pressure [kPa]	950.0	949.0	949.0	949.0
Molar Flow [kgmole/h]	1294	150.0	0.0000	1444
Mass Flow [kg/h]	3.732e+004	3263	0.0000	4.058e+004
Std Ideal Liq Vol Flow [m3/h]	43.14	8.002	0.0000	48.72
Molar Enthalpy [kJ/kgmole]	4232	-1.379e+005	-1.054e+004	-1.054e+004
Molar Entropy [kJ/kgmole-C]	144.7	167.3	213.6	213.6
Heat Flow [kJ/h]	5.475e+006	-2.068e+007	0.0000	-1.521e+007

Figure 19: Operating conditions of combustion chamber

- **Expansion Section:**

Modifications that are being made to the existing model up till now are aimed to achieve maximum power output and efficiencies in this section. The energized feed T a pressure 1240KPa is being expanded by ratio of 12:1 to get power output. Design and stream connections are shown in figure 20.

Name	To KGT101	Exhaust 1
Vapour	1.0000	1.0000
Temperature [C]	1412	1044
Pressure [kPa]	949.0	200.0
Molar Flow [kgmole/h]	2140	2140
Mass Flow [kg/h]	6.068e+004	6.068e+004
Std Ideal Liq Vol Flow [m3/h]	71.95	71.95
Molar Enthalpy [kJ/kgmole]	-5729	-1.959e+004
Molar Entropy [kJ/kgmole-C]	199.3	203.0
Heat Flow [kJ/h]	-1.226e+007	-4.192e+007

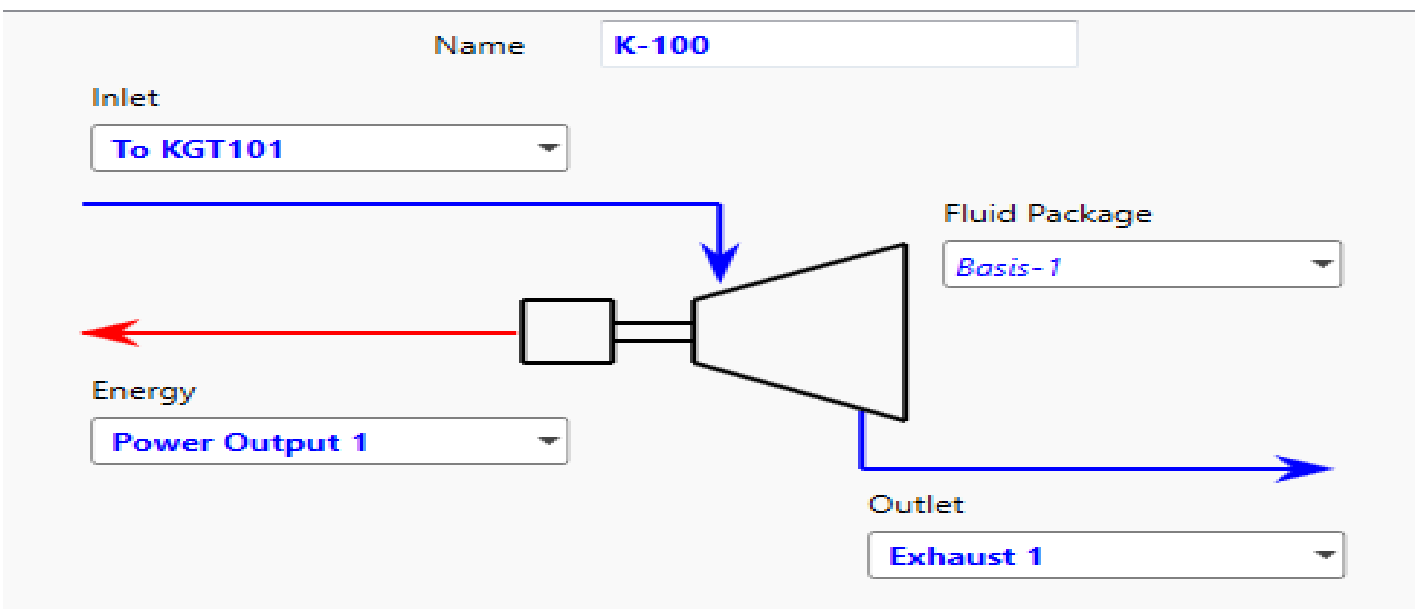
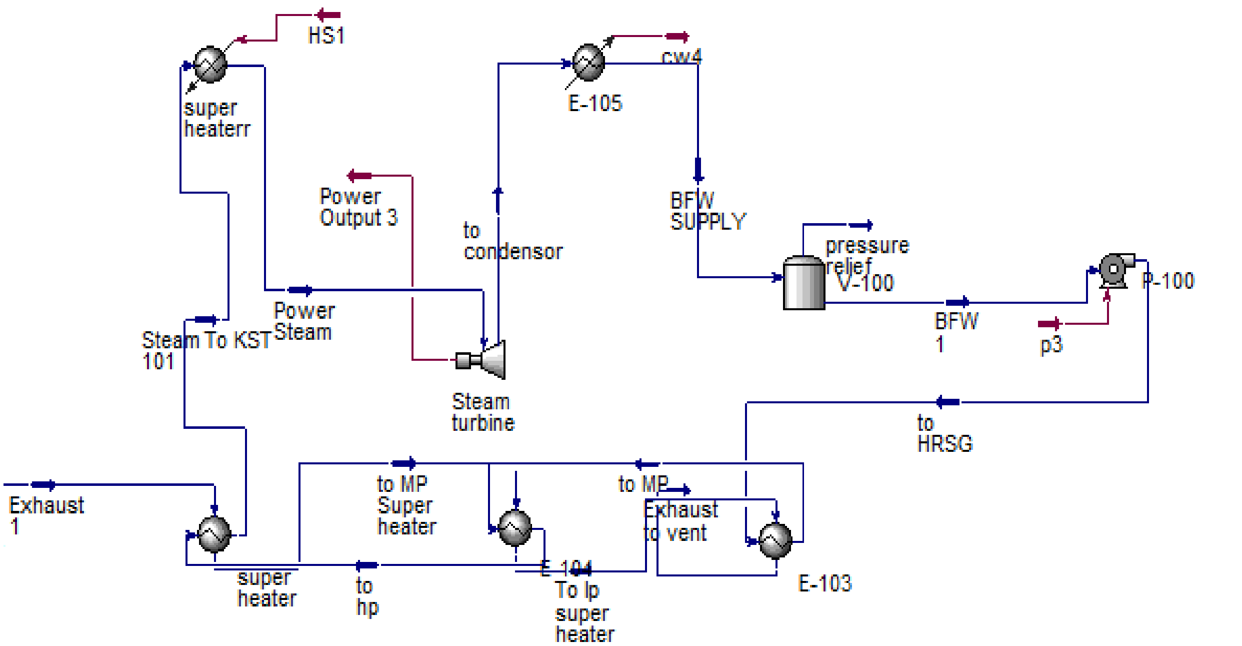


Figure 20: Operating conditions of the streams of turbine

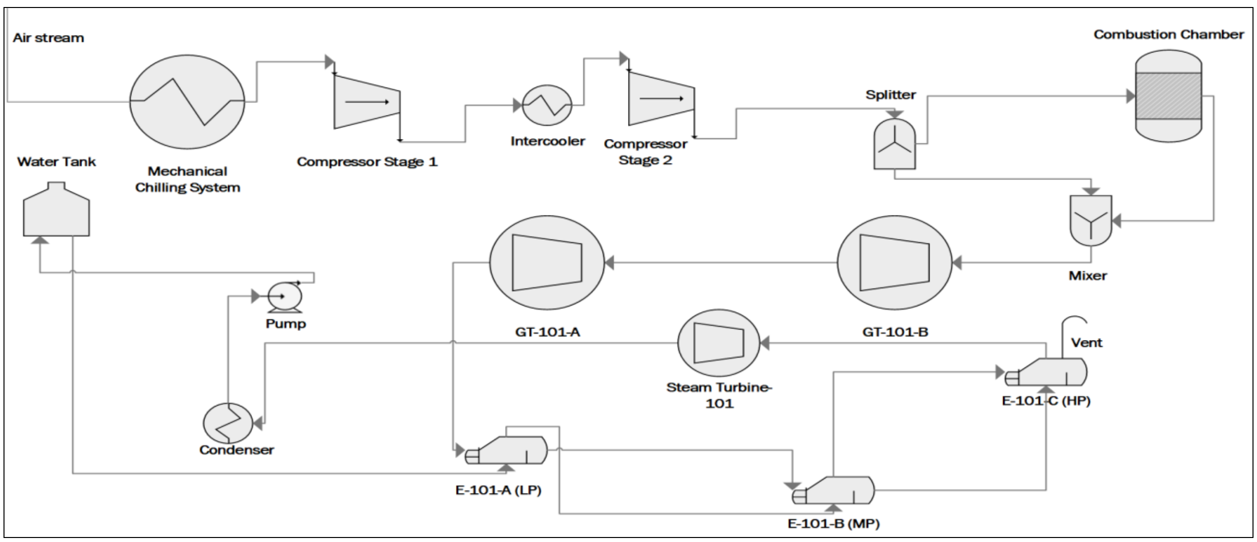
- **Heat Recovery steam generation section:**

Temperature of the exhaust stream is above 1200C and be utilized to generate 1.28megawatts of additional power by using a steam turbine. This is done by interconnecting 4 heat exchangers in a series that super heat the stream and recover energy from the exhaust. Network design is shown in figure 20.



Chapter 4 Process Description

4.1 Process Flow Diagram



The system incorporated here is of combined power cycle where the energy is generated from gas turbine operation and heat recovery steam generation section.

4.2 HRSG

A combined cycle power plant (CCPP) is a power plant where electricity production is done from the combination of both gas and steam turbines. Previous work has been done on rotor-blades' profile influence on a gas compressor effectiveness. The study stated that in the gas turbine, atmospheric air is drawn in through an intake duct and compressed to a high pressure in the compressor consisting of a cascade of several stages of blades located on a single axle in

radial form. The high-pressure air is delivered to the combustor where fuel is sprayed and the high pressure and temperature product of combustion impinges on the turbine blades which rotates the shaft that drives an electrical generator to produce electricity and the exhaust gas energy from the gas turbine is used to generate steam in a heat recovery steam generator (HRSG) which then produces more electricity from a steam turbine. There is a great potential for increased power output and efficiency, as well as reduction of emitted pollutants to the environment, in energy converting devices by recovering waste heat to a greater extent.
It was observed that combustion turbine performance has the primary impact on combined cycle plant efficiency and the next most important piece of equipment that impacts efficiency is the heat recovery steam generator (HRSG).

Tube Side In	Molar Flowrate kg/hr	Chiller	Shell Side In	Molar flowrate kg/hr

Chapter 5 Material Balance

Table 1: Overall Balance

Overall Balance			
Inlet Streams	**Molar Flowrate kg/hr**	**Outlet Stream**	**Molar flowrate kg/hr**
Air in	1320	Vent	1356
Fuel in	36		
Total	1356	Total	1356

Table 2: Mechanical Chilling System

Splitter			
Stream In	**Molar Flow rate kg/hr**	**Stream Out**	**Molar flowrate kg/hr**
Inlet air	1320	Exhaust air	462
		To combustor	858
Total	1320	Total	1320

Table 3: Splitter

CHILLER

C

h

a

Combustion Reactor Balance			
Stream In	**Molar Flowrate kg/hr**	**Stream Out**	**Molar Flowrate kg/hr**
Inlet air	1320	Power Gas	1356
Inlet fuel	36	Waste	0
Total	1356	Total	1356

pter 6 Energy *Table 4: Combustion Chamber* Balance

Compressor Section				
Stream	**Molar flowrate kgmole/hr**	**Cp kJ/kgmolK**	**Tin °C**	**Tout °C**
Air	1990	29.1	15	338.3
Q(air)kJ/hr	**1.901E+07**			
Work in KW	**5280.979083**			

Stream	**Temp °C**	**Flowrate kgm/hr**	**Cp kJ/kgK**
Chilled Water out	16	?	77.77809553
ΔHair=ΔH20 KJ/hr	1.07E+07		
Chilled water flowrate kg/hr	1154		

Table 5: Chiller Energy Balance

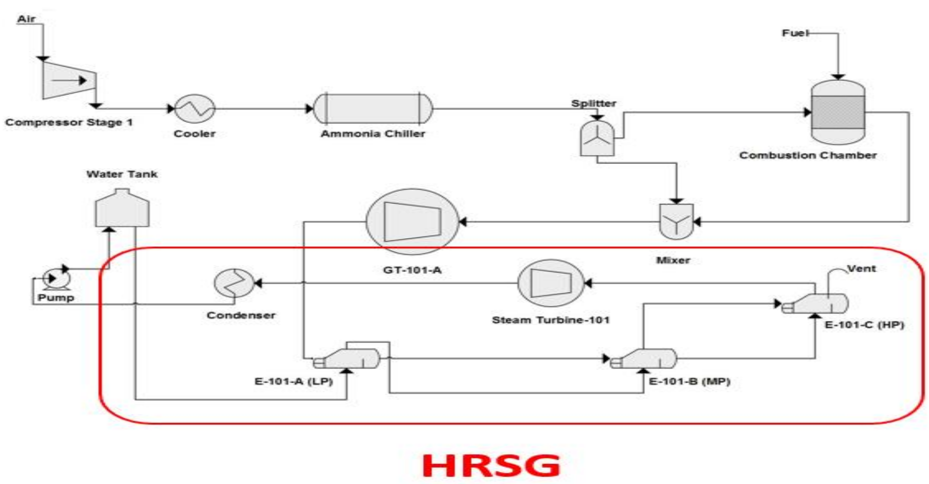
Table 6: Compression Section Load

Combustor Chamber				
Inlet Stream	**Molar Flow rate kgmol/hr**	**Cp kJ/kgK**	**Enthalpy Energy kJ**	**Temp K**
Air	1320	30.25		442
Fuel	36	37.65		304
Exhaust	1356	44.16	?	2160
LHV OF FUEL			7.48E+07	
Q reaction			**1.88E+08**	

Table 7: Heat added in Combustion Chamber

Turbine					
Stream	**Molar flowrate kg/hr**	**Cp kJ/kgmolK**	**Tin ^{o}C**	**Tout ^{o}C**	**Cp kJ/kgmolK**
Air	2140	34.63	1412	931.2	32.33
Q(air)kJ/hr	**3.421E+07**				
Work in KW	**9.5E+03**				

Table 8: Work Generated in Turbine



Heat Recovery Section			
Stream	**Temp C^o Tref 275.5**	**Flowrate kgmole/hr**	**Cp kJ/kgmolK**
Air in	1044	2140	34.2
Air out	200	2140	30.4
Water in	25	?	77.71
Water out	180	?	35.5
∆Hair=∆H20 KJ/hr	**6.34E+07**	Latent heat of Vap. kJ/kgmol	4.10E+04
H2O molar flowrate kg/hr	**1394.652602**		

Table 9: Steam Generation in HRSG section

Chapter 7 Design:

7.1 Introduction

An efficient design of heat exchangers is the mechanical chilling system is the key to our objecting of cooling down the ambient air temperature from 32 $^{\circ}$C to 15 $^{\circ}$C. These equipments were taken into consideration for design.

- Condenser from Absorption Cycle to produce liquified refrigerant
- Heat Exchanger to cool down the ambient air temperature

7.2 Types of exchangers:

7.2.1 Shell and tube heat exchangers

A shell and tube heat exchanger consists of a cylindrical shell with tube bundles passing through it. Fluid in the tubes is called tube side fluid, while the fluid in the shell is called shell side fluid. Preferably, hot fluid is kept in the tubes to reduce the heat lost to the environment. Shell fluid may flow co-currently, counter currently or cross to the tube side fluid.

7.2.2 Plate type heat exchangers

A plate heat exchanger has series of plates or fins that give direction to the fluid to pass through. Second fluid passes in the channels made by the alternate plates.

7.3 Condenser Design

Condenser type: Shell and tube

Tube-side fluid: Cooling water

Shell -side fluid: R-134a refrigerant

7.3.1 Material Selection

Shell is containing water as the cooling agent, hence common Carbon Steel can be used to manufacture shell because of its low maintenance costs.

Material selected for tubes: Stainless steel 904L

7.3.2 Design Calculations

Step 1. Extract fluid data

$\dot{m}_{cooling\ water} = 5.77\ kg/s$

$\dot{m}_{Refrigerant} = 0.67\ kg/s$

$\dot{Q} = 149.58\ kJ/s$

$T_1 = 22\ °C$

$T_2 = 28\ °C$

$t_1 = 0.4372\ °C$

$t_2 = -17.78\ °C$

Step 2. Collect the values of physical property

R134a Refrigerant:

$\mu_t = 1..58\ \times 10^{-4} kg/m\ s$

$k_t = 0.5903\ W\ m/°C$

$cp = 1.04\ J/g°C$

Water:

$\mu_t = 1.02\ \times 10^{-3} kg/m\ s$

$k_t = 0.5903\ W\ m/°C$

$cp = 4.187\ J/g°C$

Step 3. Select exchanger type and fluid location

As mentioned previously,

Exchanger type: Shell and tube heat exchanger

Fluid location: Tube side fluid: Cooling water

Shell side fluid: R134a refrigerant

Step 4. Select an approximate value for U

$U_{approx} = 300\ W/m^2\ °C$

Step 5. Calculate LMTD and temperature correction factor

$$LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{ln(\frac{T_1 - t_2}{T_2 - t_1})}$$

$$LMTD = 34.16\ °C$$

For correction factor,

$$R = \frac{(T_1 - T_2)}{(t_2 - t_1)}$$

$$R = 0.33$$

$$S = \frac{t_2 - t_1}{(T_1 - t_1)}$$

$$S = 0.84$$

$$F_t = 0.98$$

Step 6. Calculate preliminary area of the exchanger

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

$$A = \frac{Q}{U \times LMTD \times F_t}$$

$$A = 14.60\ m^2$$

Step 7. Choose shell and tubes specifications

Tube: Material: SS-904L

$d_{to} = 20$ mm

$d_{ti} = 16$ mm

L = 3.66 m

$$n_t = \frac{A}{\pi \times d_{to} \times L}$$

$n_t = 64$

Shell:

$D_s = 325.63$ mm

$N_p = 1$

2 tube passes

Pitch: Triangular because of the non-fouling nature of fluids

$p_t = d_{to} \times 1.25$

Segmental baffles

Baffle spacing = 0.5 D_s = 162.87 mm

Step 8. Calculate the individual heat transfer coefficients for each fluid

Tube fluid (Cooling water):

$$Re = \frac{G_t \times d}{\mu}$$

$$Re = 14{,}067$$

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

$$Pr = 7.23$$

$$\frac{\mu}{\mu_w} = 1$$

Using equation:

$$Nu = C \times (Re)^{0.8} \times (Pr)^{0.33} \times (\frac{\mu}{\mu_w})^{0.14}$$

$$Nu = 106.08$$

$$Nu = \frac{h_t \times d_e}{k}$$

$$h_t = \frac{Nu \times d_e}{k}$$

$$h_t = 3129.43\ W/m^2\ °C$$

Shell fluid (R134a Refrigerant):

$$A_s = \frac{p_t - d_{to} \times D_s \times l_B}{p_t}$$

$$A_s = 0.0106\ m^2$$

$$u_s = \frac{G_s}{\rho}$$

$$u_s = 0.06971\ m/s$$

$$d_e = \frac{1.10}{d_o} \times (p_t^2 - (0.917 \times d_{to}^2))$$

$$d_e = 14.20\ mm$$

$$Re = \frac{G_s \times d_e}{\mu}$$

$$Re = 5717.50$$

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

$$Pr = 15.57$$

Correction factor: $j_h = 3.8 \times 10^{-3}$

$$Nu = j_h \times Re \times (Pr)^{0.33} \times (\frac{\mu}{\mu_w})^{0.14}$$

$$Nu = 53.75$$

$$Nu = \frac{h_s \times d_e}{k}$$

$$h_s = \frac{Nu \times d_e}{k}$$

$$h_s = 399\ W/m^2\ °C$$

Step 9. Calculate U and find error (with respect to U_{approx})

$$U = \frac{1}{h_s} + \frac{1}{h_{sd}} + \frac{d_s ln(\frac{d_s}{d_t})}{2 \times k_w} + \frac{d_s}{d_t} \times \frac{1}{h_{td}} + \frac{d_s}{d_t} \times \frac{1}{h_t}$$

$$h_{sd} = 5000\ W/m^2\ °C$$

$$h_{id} = 5000\ W/m^2\ °C$$

$$k_w = 16.0\ W/m°C$$

$$U = 286.13\ W/m^2\ °C$$

$$Error = \frac{U_{approx} - U_{calc}}{U_{approx}} \times 100$$

$$Error = 4.62\ \%$$

Step 10. Calculate the pressure drop

Tube side:

$$\Delta P_t = N_p \times \left[(8 \times j_f \times (\frac{L'}{d_t}) \times (\frac{\mu}{\mu_w})^{0.14}) + 2.5 \right] \times \frac{\rho \times u_t^2}{2}$$

$\Delta P_t = 7.75\ kPa$

Shell side:

$$\Delta P_s = 8 \times j_f \times (\frac{D_s}{d_e}) \times (\frac{L}{l_B}) \times (\frac{\rho \times u_s^2}{2}) \times (\frac{\mu}{\mu_w})^{-0.14}$$

$\Delta P_s = 0.270\ kPa$

7.4 Heat Exchanger Design

Condenser type: Shell and tube

Tube-side fluid: Chilled Water water

Shell -side fluid: Inlet Air

7.4.1 Material Selection

Shell is containing air coming at 32 °C, therefore stainless Steel can be used to manufacture shell because of its low maintenance costs.

Material selected for tubes: Stainless steel 904L

7.4.2 Design Calculations

Step 1. Extract fluid data

$\dot{m}_{air} = 16.03\ kg/s$

$\dot{m}_{chilled\ H_2O} = 5.77\ kg/s$

$\dot{Q} = 177.80\ kJ/s$

$T_1 = 32\ °C$

$T_2 = 15\ °C$

$t_1 = 5\ °C$

$t_2 = 16\ °C$

Step 2. Collect the values of physical property

Ambient Air:

$\mu_t = 1.91\ \times 10^{-5} kg/m\ s$

$k_t = 0.0264\ W/m°C$

$cp = 1.005\ J/g°C$

Chilled Water:

$\mu_t = 1.02 \times 10^{-3} kg/m\ s$

$k_t = 0.578\ W/m°C$

$cp = 4.187\ J/g°C$

Step 3. Select exchanger type and fluid location

As mentioned previously,

Exchanger type: Shell and tube heat exchanger

Fluid location: Tube side fluid: Chilled Water

Shell side fluid: Inlet Air

Step 4. Select an approximate value for U

$U_{approx} = 800\ W/m^2\ °C$

Step 5. Calculate LMTD and temperature correction factor

$$LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{ln(\frac{T_1 - t_2}{T_2 - t_1})}$$

$LMTD = 12.77\ °C$

For correction factor,

$$R = \frac{(T_1 - T_2)}{(t_2 - t_1)}$$

$R = 1.55$

$$S = \frac{t_2 - t_1}{(T_1 - t_1)}$$

$S = 0.41$

$F_t = 0.76$

Step 6. Calculate preliminary area of the exchanger

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

$$A = \frac{Q}{U \times LMTD \times F_t}$$

$A = 22.88\ m^2$

Step 7. Choose shell and tubes specifications

Tube: Material: SS-904L

$d_{to} = 20 \text{ mm}$

$d_{ti} = 16 \text{ mm}$

$L = 4.0 \text{ m}$

$$n_t = \frac{A}{\pi \times d_{to} \times L}$$

$n_t = 92$

Shell:

$D_s = 0.370 \text{ m}$

$N_p = 1$

2 tube passes

Pitch: Triangular because of the non-fouling nature of fluids

$p_t = d_{to} \times 1.25$

Segmental baffles

Baffle spacing = 0.5 D_s = 0.185 m

Step 8. Calculate the individual heat transfer coefficients for each fluid

Tube fluid (Chilled Water):

$$Re = \frac{G_t \times d}{\mu}$$

$Re = 9782$

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

$Pr = 7.37$

$$\frac{\mu}{\mu_w} = 1$$

Using equation

$$Nu = j_h \times (Re) \times (Pr)^{0.33} \times (\frac{\mu}{\mu_w})^{0.14}$$

$$Nu = 73.74$$

$$Nu = \frac{h_t \times d_e}{k}$$

$$h_t = \frac{Nu \times d_e}{k}$$

$$h_t = 2664.72\ W/m^2\ °C$$

Fouling factor

$$j_h = 0.0039$$

Shell fluid (Inlet Air):

$$A_s = \frac{p_t - d_{to} \times D_s \times l_B}{p_t}$$

$$\mathrm{A_s} = 0.01366\ \mathrm{m^2}$$

$$u_s = \frac{G_s}{\rho}$$

$$u_s = 1.36\ m/s$$

$$d_e = \frac{1.10}{d_o} \times (p_t^2 - (0.917 \times d_{to}^2))$$

$$d_e = 0.0142\ m$$

$$Re = \frac{G_s \times d_e}{\mu}$$

$$Re = 8.72 \times 10^6$$

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

$$Pr = 0.7271$$

Correction factor: $j_h = 3 \times 10^{-3}$

$$Nu = j_h \times Re \times (Pr)^{0.33} \times (\frac{\mu}{\mu_w})^{0.14}$$

$$Nu = 23548$$

$$Nu = \frac{h_s \times d_e}{k}$$

$$h_s = \frac{Nu \times d_e}{k}$$

$$h_s = 5549.42\ W/m^2\ °C$$

Step 9. Calculate U and find error (with respect to $U_{approx)}$

$$U = \frac{1}{h_s} + \frac{1}{h_{sd}} + \frac{d_s ln(\frac{d_s}{d_t})}{2 \times k_w} + \frac{d_s}{d_t} \times \frac{1}{h_{td}} + \frac{d_s}{d_t} \times \frac{1}{h_t}$$

$$h_{sd} = 7500\ W/m^2\ °C$$

$$h_{id} = 5000\ W/m^2\ °C$$

$$k_w = 50\ W/m\ °C$$

$$U = 928.3\ W/m^2\ °C$$

$$Error = \frac{U_{approx} - U_{calc}}{U_{approx}} \times 100$$

$$Error = 16.0\ \%$$

Step 10. Calculate the pressure drop

Tube side:

$$\Delta P_t = N_p \times \left[(8 \times j_f \times (\frac{L'}{d_t}) \times (\frac{\mu}{\mu_w})^{0.14}) + 2.5 \right] \times \frac{\rho \times u_t^2}{2}$$

$$\Delta P_t = 4.01\ kPa$$

Shell side:

$$\Delta P_s = 8 \times j_f \times (\frac{D_s}{d_e}) \times (\frac{L}{l_B}) \times (\frac{\rho \times u_s^2}{2}) \times (\frac{\mu}{\mu_w})^{-0.14}$$

$$\Delta P_s = 15.4\ kPa$$

Chapter 8: ECONOMIC ANALYSIS

8.1 Costs of Equipment

Table 6: Purchase Cost of Equipment (C) = PF x MF x TF x CF

PF	Pressure Factor
MF	Material Factor
TF	Tube Factor
CF	Base Cost of Equipment

Table 7: Correlations Used

Base Cost	exp{11.147-0.9186[ln(Area)]+0.09790[ln(Area)]^2}	A(ft^2)
Material Factor	a+[(Area)/100]^b	A(ft^2)
Pressure Factor	0.9803+0.018(P/100)+0.0017(P/100)^2	P(psia)

Table 8: Purchase Cost of Chilling System Heat Exchangers

Evaporator	**Cost ($)**
Base Cost	13495.67
Material Factor	3.76
Pressure Factor	2.23
Tube Factor	1
Total Purchase Cost	113090.87

Condenser	**Cost ($)**
Base Cost	12898.76
Material Factor	3.78

Pressure Factor	2.23
Tube Factor	1
Total Purchase Cost	108590.49

Table 9: Purchase Cost of HRSG

E-101	**Cost ($)**
Base Cost	13726.19
Material Factor	3.76
Pressure Factor	1.98
Tube Factor	1
Total Purchase Cost	102105.43

E-102	**Cost ($)**
Base Cost	13956.28
Material Factor	3.75
Pressure Factor	1.98
Tube Factor	1
Total Purchase Cost	103304.56

E-103	**Cost ($)**
Base Cost	14853.92
Material Factor	3.73

Pressure Factor	1.97
Tube Factor	1
Total Purchase Cost	109308.56

8.2 Fixed Capital Cost

Table 10: Physical Equipment Cost = Equipment Cost x (1+f1+f2+f3)

Piping Factor	f1=0.5
Instrumentation Factor	f2=0.2
Electrical Factor	f3=0.1
Attainment	0.98

Component	**Equipment Cost ($)**	**Physical Equipment ($)**
Evaporator	113,090.81	203,563.45
Condenser	108,590.49	195,462.88
HRSG	314,718.52	566,493.33

8.3 Operating Cost Annually

Table 11: Calculation Correlations

Physical Equipment Cost ($)	Sum of Evaporator, Condenser, HRSG
Maintenance Cost ($)	5% of Physical Equipment Cost
Miscellaneous Cost ($)	1% of Maintenance Cost
Total Operating Cost ($)	Maintenance Cost + Miscellaneous Cost

Annual Power (kW)	2,200 x 24 x 365 x Attainment (0.98)
Annual Power ($)	16 cents per kilo watt produced
Annual Profit ($)	Annual Power – Total Operating Cost
Breakeven	100/[(Profit/Physical Equip. Cost)*100]

Table 12: Annual Cost and Power Outputs

Physical Equipment ($)	965,519.66
Maintenance Cost ($)	48,275.98
Miscellaneous Cost ($)	4,827.60
Total Operating Cost ($)	53,103.58
Power Generated (kW)	2,200.00
Annual Power (kW)	18,886,560.00
Annual Power ($)	2,968,418.07
Profit ($)	2,915,315.49
Breakeven	0.34

Chapter 9: HAZOP ANALYSIS

The term risk refers to the negative influence of an event or an action on a planned procedure. (1) This can include a business procedure such as a production procedure or even a technical procedure such as combustion in a gas turbine and its operability. A risk can produce especially serious safety problems especially in the context of technical procedures. This can also result in very high levels of economic damage. This becomes clear using the example of aerospace engineering. A risk and the resulting damage can result in danger for life and limb, but also result in considerable financial consequences. As a result, and due to the ever-complexes composition of our industrialized world with its high technical standards, risk management has become indispensable. The earlier (for example during product development) a risk is recognized and reduced, the more successful is a product or procedure (2). Risk management includes all measures for the recognition, analysis, evaluation, monitoring and control of risks. In 2005, ISO decided to develop a risk management standard. ISO/DIS 31000 Risk Management (Figure 1) is subdivided into the three sections Principles; Risk management framework; and Risk management process.

The principles for risk management are:

- It creates values
- It is an integrated part of organizational processes
- It is part of the decision-making process
- It deals expressly with uncertainty
- It is systematic, structured and up-to-date
- It is based upon the best available information
- It is tailored
- It takes into account human and cultural factors
- It is transparent and comprehensive
- It is dynamic, iterative and reacts to changes
- It facilitates continual improvement and organizational strengthening

HAZOP Analysis

Hazard and Operability Analysis, more commonly known as HAZOP analysis is a systematic way of identifying possible hazards in any process. In this analysis, the process is fragmented into steps and each step is analyzed to see what could go wrong. It's an essential step towards safeguarding the working personnel, the equipment, and the efficiency of the operation.

Table 13: HAZOP Guide Words and Meanings

Guide Word	Meaning
No	Negation of design intent
Less	Quantitative decrease
More	Quantitative increase
Part of	Qualitative decrease
Reverse	Opposite of the intent
Other than	Complete substitution
As well as	Qualitative increase

HAZOP for Gas Turbine

Table 14: HAZOP for Gas Turbine with reference to flow

Deviation	Causes	Consequences	Recommendation
Less	Filter is partially blocked	Reduced efficiency of gas turbine	Filter needs to be cleaned
More	Filter has a leakage	Impurities	Proper sealing of

			leaked filter
No	Blockage	Shut down	Get rid of blockage

Table 15: HAZOP for gas turbine with reference to pressure

Deviation	Causes	Consequences	Recommendation
Less	Deteriorated compressor blades	Reduced power output	Replace blades
No	Blockage	Shut down	Get rid of blockage

Table 16: HAZOP for gas turbine with reference to temperature

Deviation	Causes	Consequences	Recommendation
Less	Not enough fuel	Reduced exhaust temperature	Thoroughly examine the fuel system
More	Fuel more than required	More chances of damage/wear on gas turbine blades	Thoroughly examine the fuel system

Chapter 10: Concluding Remarks

`10.1 Power Output

Efficiency and power output are the key indexes of measuring gas turbine performance and its variation with the fluctuations of inlet air temperature. Normally, gas turbine power output and efficiency increase when the inlet air temperature is decreasing. The power required to drive the compressor decreases with the decrease of ambient temperature. In the T-S diagram it can be seen that the gap between two constant pressure lines reduces with the decreasing temperature. Hence the power load in the compressor decreases with the decrease of ambient temperature.
It can be clearly noticed that the actual power responds to a temperature change more than the theoretically derived power variation. This difference could be due to the increase of mechanical losses with the increase of temperature. Most probably it could be the heat loss which is higher when the temperature is higher, causing the observed difference.

10.2 Efficiency

Theoretical efficiency was calculated using the equation. It is the equation for efficiency of the real Brayton cycle. According to the equation, ambient temperature, pressure ratio, turbine inlet temperature and ratio of specific heats are the factors affecting the efficiency. In theoretical efficiency calculation it is assumed that only the ambient temperature is varying with the charge air cooling and all the other factors are constant. From the heat rate test results presented in previous chapter, it is derived that the efficiency at 33^0 C is 21.26%. Hence, we can shift the graph for efficiency to fit the actual interpretation of the efficiency variation with the ambient temperature as shown in Figure 6.2.

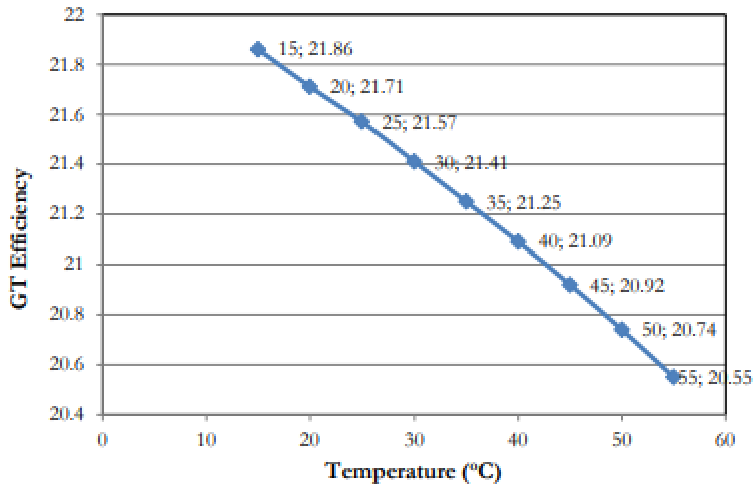


Figure 21;Efficiency variation with ambient air temperature

10.3 Feasibility

According to studies, 27^0 C is the average air temperature in Pak Arab year-round, and bringing it down to ISO value of 15^0 C gives Rs.5.9Mn of annual savings for the gas turbine unit at the established operational hours. This saving is only from the decreased amount of fuel due to efficiency improvement. The annual power generation by the Pak Arab plant can be increased by more than 20%.

10.4 Future Developments

Further studies can be done on this project to increase the accuracy of feasibility assessment and to enhance the economic benefits. One option is to cool the inlet air for all five gas turbines installed at the Pak Arab power station by using exhaust gas from one gas turbine. This method can keep cost down while achieving higher economic feasibility. It will require a chiller system with larger cooling capacity which should be enough to cool the air for all the five gas turbines in the power station. The exhaust gas from a single turbine does have the energy content to deliver such large cooling load, however, that particular machine should be kept operating with priority in order to supply precooled air to all other operating turbines at any particular moment. Again, the overall increase of power production for the entire plant should also be valorized for the financial benefits according to the actual operational hours for each of the available gas turbines in the plant.

References

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