UNIVERSITY OF TURKISH AERNAUTICAL ASSOCIATION
INSTITUTE OF SCIENCE AND TECHNOLOGY

EVALUATION THE IMPACT OF USING DIFFERENT COOLING TECHNOLOGIES ON POWER BOOSTING OF THE GAS TURBINE UNIT IN DUHOK POWER STATION – IRAQ

Master Thesis

Rabeea AL-AFFAS

Institute of Science and Technology

Mechanical and Aeronautical Engineering Department

AUGUST, 2016

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Tez Danışmanı : **Doç. Dr. Murat DEMİRAL**
Türk Hava Kurumu Üniversitesi ....

Jüri Üyeleri : **Doç. Dr. Murat DEMİRAL**
Türk Hava Kurumu Üniversitesi ....

: **Yrd. Doç. Dr. Mohamed ELMNEFI**
Türk Hava Kurumu Üniversitesi ....

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Rabeea, Al-AFFAS

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AL-AFFAS, Rabeea

M.Sc., Department of Mechanical Engineering

Supervisor: Assist. Prof. Dr. Murat DEMIRAL

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Peak demand for electric power in Iraq occurs almost during all the days of Summer and is almost double the off-peak demand. The demand profile is ill-matched to the performance profile of combustion turbines as their power output decreases with increased inlet-air temperature. Approximately 55%of the annual energy generated in Iraq is done by combustion turbines, in the meanwhile turbines loss a 46%from their capacity during the Summer due to ambient air temperatures up to 50C°. Industrial gas turbines that operate at constant speed are constant-volume-flow combustion machines. As the specific volume of air is directly proportional to the temperature, the increase in the air density lead to higher air mass flow rate, once the volumetric rate is constant. Consequently, the gas turbine power output enhances. A gas turbine inlet air cooling technique is a useful option for increasing output. Inlet air cooling increases the power output by taking advantage of the gas turbine's feature of higher mass flow rate, due the compressor inlet temperature decays. Different methods are available for reducing compressor intake air temperature. In this work, the influence of integration two type of cooling system, the first is evaporative cooling, which make utilization of the dissipation of water to decrease the gas turbine bay air temperature and the second one is absorption cooling system, which make use of exhausted hot gases to cool the compressor air intake through a warmth exchanger situated in the duct inlet to expel the warmth from the channel air, at Duhok Gas Power Station, Erbil, Iraq is presented. A thermodynamic examination of gas turbine execution is completed to compute power yield and warm proficiency at various channel air temperature and weight proportion conditions. The outcomes got with this model are contrasted and the estimations of the condition without cooling in this named of Base-Case. The obtained results showed that the evaporative cooling system is capable of boosting the power and enhancing the efficiency of the studied gas turbine unit in a way much cheaper than absorption cooling coil system but associated with a limited cooling potential by wet bulb temperature. On the other hand, the absorption chiller provides the highest increment in annual energy generation with higher unit energy costs.

Keywords: Gas turbine, Turbine inlet cooling, TIC, Evaporative cooling, Chiller absorption.

ÖZET

IRAK, DUHOK ENERJİ İSTASYONUNDA GAZ TÜRBİNİ BİRİMİNDEKİ ENERJİYİ ATRITMA AMACIYLA YAPILAN FARKLI SOĞUTMA TEKNİKLERİN ETKİSİ

AL-AFFAS, Rabeea
Yüksek Lisans, Makine Mühendisliği Anabilim Dalı
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Irak'ta elektrik enerjisine olan talep yaz günleri dışında bir birim ise yaz günlerinde hemen hemen bunun iki katı olmaktadır. Türbinlerin enerji çıktısı arttırılıp hava girişi azaldığında gaz türbinlerinin performans profilleri bozulmaktadır. Irakta üretilen yıllık enerjinin %55'i gaz türbinleriyle yapılmaktadır. Ancak bölgesel hava sıcaklık değerlerinin 50°C'ye ulaşmasından ötürü yaz boyunca türbinler kapasitelerinin % 46'sını kaybederler. Türbinler sabit hızda ve sabit hacim akışlı kombüsyon makinalarıdır. Havanın spesifik hacmi doğrudan sıcaklıkla doğru orantılı olduğundan volumetrik artış sabitken hava yoğunluğundaki artış hava akış debisinin artışına yol açmaktadır. Bu da gaz türbinlerinde enerji üretimini arttırmaktadır. Böyle bir durumda gaz türbin girişinin hava soğutma tekniği enerji verimliliği açısından önemlidir. Kompresör giriş sıcaklığının düşmesiyle birlikte hava soğutma girişi gaz türbininin daha yüksek hava akış debisine ulaşmasını sağlayarak enerji üretimi arttırmaktadır. Kompresör giriş hava sıcaklığını düşürmek için farklı metotlar kullanılır. Bu çalışmada iki soğutma tekniğinin entegre etkisi incelenecektir. İlk olarak gaz türbin kanal sıcaklığını düşürmek için su kullanılmasıdır. Diğer teknik ise kanalda oluşan ısıyı azaltmak için giriş kanalına yerleştirilen ısı eşanjörü aracılığıyla kompresör hava girişini soğutmaktır. Bu da sistem içinde tüketilmiş sıcak gazların kullanımıyla soğutma emme sistemiyle

mümkündür. Erbil Irak, Duhok Gaz Enerji İstasyonu bu sisteme örneklik teşkil etmektedir. Gaz türbini uygulamasının termodinamik incelenmesiyle enerji üretimi, kanal hava akış derecesi ve ağırlık oranı koşullarının hesaplanması amaçlanmaktadır. Bu modellerle elde edilen veriler ile temel durum olarak adlandırılan soğutmasız koşul tahminleri karşılaştırılmaktadır. Elde edilen veriler buharlaştırıcı soğutucu sistemin enerjiyi yükseltebilmesini ve emici soğutucu bobin sisteminden daha ucuz bir yolla gaz türbinlerinin verimliliğinin arttığı ortaya çıkmaktadır ancak buna bağlı olarak ıslak cıva haznesi sıcaklığının soğutma potansiyeliyle ilişkisinin sınırlı olduğu gözlenmiştir. Diğer taraftan emici soğutucu sistemin daha maliyetli olduğu ve bunun enerji fiyatlarına yansıyarak yıllık enerji maliyetlerinde yüksek artışa sebep olacağı açıktır.

Anahtar Kelimeler: Gaz tirbünü, soğutma tirbünü girişi, Buharlaşmalı soğutma, Chiller emilimi.

LIST OF ABBREVIATIONS

ISO	:	International Standards Organization
Cp	:	Heat Capacity at Constant Pressure (kJ/kg K)
g	:	Acceleration due to Gravity (m/s^2)
LHV	:	Lower Heating Value (kJ/Kg)
$\dot{m}$	:	Mass Flow Rate (Kg/s)
R	:	Universal Constant (kJ/Kg K)
P	:	Pressure (kPa)
P_r	:	Pressure ratio
ΔP	:	Pressure drop (kPa)
r_p	:	Pressure Ratio
s	:	Specific Entropy (kJ/Kg K)
T	:	Temperature (K)
DGPS	:	Duhok Gas Power Station
MoE	:	Iraqi Ministry of Electricity
ρ	:	Density of the air
$\dot{v}$	:	Volumetric air flow rate
rpm	:	Turbine speed
W_C	:	Compressor Work
W_{GT}	:	Power output of ideal cycle
W_T	:	Turbine expansion power
λ	:	Coefficient of useful power
τ	:	Temperature ratio
k	:	Adiabatic index
ω	:	Air specific humidity
ε	:	Evaporative cooling effectiveness
T_{DB}	:	Dry-bulb temperature of the air
T_{WB}	:	Wet-bulb temperature of the air
RH	:	Relative humidity
q_s	:	Sensible heat
q_l	:	Latent heat
h_{fg}	:	Latent heat of water at Wet-bulb temperature
η_c	:	Compressor isentropic efficiency
γ	:	Specific heat ratio
η_T	:	Turbine isentropic efficiency
$\dot{E}$	:	Energy flow rate
BC	:	Base case operation without adding any cooling system
EC	:	Operation with adding evaporative cooling system
AS	:	Operation with adding absorption cooling system

CHAPTER ONE

INTRODUCTION

1.1 Introduction

Gas turbine engine is an internal combustion engine, it comprise of a compressor to raise ignition gaseous tension, a burning chamber where the fuel/air blending is burned, and a turbine that through development remove vitality from the combustion gases. These cycles works as per the open Brayton thermodynamic cycle and display low warm effectiveness and are alluded as burning turbines [1].

Gas turbines have increased boundless acknowledgment in the power generation, mechanical drive, and gas transmission markets. Their minimization, high energy to-weight proportion, and simplicity of establishment have made them a prominent prime mover [3]. It have been utilized for power era as a part of most nations around the globe. Previously, their utilization has been for the most part constrained to creating power in times of top power request. Gas turbines are perfect for this application as they can be begun and halted rapidly empowering them to be conveyed into administration as required to take care of vitality demand crests [2]. Notwithstanding, because of accessibility of fossil fuel at generally shabby costs, numerous nations around the globe, e.g. Iraq, utilize extensive ordinary GTs as base burden units. Such frameworks, particularly those working in an open or basic cycle have the disservice of being minimum proficient thus the unit expense of created power is generally high. The normal productivity of gas turbine plants in Iraq over the last two decades was the range 20-25% [6]. Such low efficiencies can be ascribed to numerous reasons, for example, operation mode, poor support, motor size and age [2]. Improvement in hot section materials, new coating, cooling technologies, and aerodynamics have allowed increases in firing temperature. Consequently, thermal efficiencies are currently very attractive, with simple cycle efficiencies ranging between 32 and 42 %and combined cycle efficiencies reaching the 60%mark [3]. The active use of natural resources in all approaches of community has become more important, as well as technological progress and growth of population has increased the need for the best kind of energy sources. These sources availability is limited, as the renew ratio is unhurried which they are regarded non-renewable [4].

1.2 Problem Statement

Usually, the rated capacities of combustion turbines are based on standard ambient air conditions, and zero inlet and exhaust pressure drops, as specified by the International Organization for Standardization (ISO). The air inlet conditions are: air temperature 15 C°, relative humidity 60 %, absolute pressure 101.325 kPa at sea level [1].

Burning turbines are steady volume motors and their energy yield is straightforwardly corresponding and constrained by the air mass stream rate entering the motor. As the compressor has a settled limit for a given rotational pace and a volumetric stream rate of air, their volumetric limit stays steady and the mass stream rate of air go into the gas turbine fluctuates with encompassing air temperature and relative dampness, however the temperature surrounding is the variable that has the best impact on the gas turbine execution. The temperature encompassing ascent results in abatement in air thickness, and therefore, in the diminishment of the mass stream rate. In this manner, less air goes through the turbine and the force yield is lessened at a given turbine passage temperature [1]. Gas turbine yield and proficiency is a solid capacity of the encompassing air temperature. Contingent upon the gas turbine sort, power yield is lessened by a rate somewhere around 5%and more than 10%of the ISO-appraised power yield (15 C°) for each 10 K increment in encompassing air temperature [3]. The work of Ibrahim shown that an increment of 1C° in the compressor air inlet temperature decreases the gas turbine power output by 1%. In the meantime, the particular warmth utilization increment by a rate somewhere around 1.5% and more than 4% [3]. The most astounding misfortunes in gas turbine influence yield as a rule agree with times of high power request. This represents a noteworthy issue for turbine administrators in the quickly deregulating power era area where the structure of supply understandings and the flow of an open business sector imply that force maker are paid fundamentally more for force created amid appeal periods [3]. In the Saudi Electric Company's (SEC), for example, approximately 42%of the annual energy is generated by combustion turbines(capacity 14 GW), and during the summer these turbines suffer a 24%decrease in their capacity (approximately 3.36 GW), due to ambient temperature up to 50 C°[7].

1.3 Thesis Objective

Gas turbine have been used for power generation in several places in the world [1], and each regions have different climatic conditions, in other words, ambient temperature and relative humidity ratio. For example, the weather in desert areas is hot and dry while in the is coastal regions weather is hot and wet. In Iraq, during the Summer season, the ambient temperature may reach as high as 47 C°, or even higher in July and August when peak demand occurred due to increasing demand on electrical energy for refrigerating, air-conditioning and cooling systems. The ambient temperature has a strongest influence on the gas turbine performance [2]. The objective of this thesis is to survey the cutting edge in applications for diminishing the gas turbine admission air temperature in gas-turbine-based force plants. Two different cooling, methods for reducing the intake air temperature are examined, direct evaporative cooling and indirect absorption cooling systems. Simple cycle (SC) gas turbine plant owned and operated by Iraqi Mass Group Holding Company have been selected as test case for investigation of the benefits from coordination of the distinctive gas turbine consumption air-cooling techniques. Taking after the discourse on the distinctive cooling techniques, the consequences of the impact of combination of the delta air-cooling framework under scrutiny on the operation of the chose power plants are exhibited and examined. The calculation are performed on a typical meteorological year (TMY) data of operation and the time-varying climatic conditions are taken into account. Finally, the economics from integrations of the different cooling systems are calculated and compared.

As a result, it intends to help Decision-creators, potential financial specialists and different experts to comprehend and assess this vitality sparing potential arrangement, which is currently accepting a lot of positive consideration, both for its vitality effectiveness and natural advantages [6].

1.4 Organization of the Thesis

Chapter one, displays general overviews about electrical power generation, effect of ambient air temperature to different types of power plants and all related concepts.

Chapter two, includes definition of gas turbine main parts, studying of close and open cycle of gas turbine, as well as the definition of Brayton cycle, deviation of actual gas turbine from idealized state, furthermore simplification of cooling systems integrations to the base-power plant.

Chapter three, this chapter gives a brief about Duhok city that contains Duhok power plant and description of existing unit GE 9E.03 Heavy Duty Gas Turbine.

Chapter four, this chapter gives a brief description of the used cooling systems, principles and equations will be included, also proposed method of work.

Chapter five, calculations and results.

Chapter Six, conclusions and future work.

CHAPTER TWO

GAS TURBINE

2.1 Historical Introduction

The modern axial-flow turbine developed from a long line of inventions stretching back in time to the aeolipile of Hero (aka Hero) of Alexandria around 120 BC. Although we regard it as a toy it did demonstrate the important principle that rotary motion could be obtained by the expansion of steam through nozzles. Over the centuries, many developments of rotary devices took place with wind and water driven mills, water driven turbines, and the early steam turbines of the Swedish engineer Carl de Laval in 1883. The main problems of the de Laval turbines arose from their enormous rotational speeds, the smallest rotors attained speeds of 26,000 rpm and the largest has peripheral speeds excess of 400 m/s. Learning from these mistakes, Sir Charles Parsons in 1891 developed a multistage (15 stage) axial-flow steam turbine, which had a power output of 100 kw at 4800 rpm. Later, and rather famously, a Parsons steam turbine rated at 1570 kw was used to power a 30 meter long ship, Turbinia, at what was regarded as an excessive speed at a grand review of naval ships at Spithead, England, in 1897 [8].

From this point, the design of steam turbines evolved rapidly, By 1920, General Electric was supplying turbines rated at 40MW for generating electricity. Significant progress has since been made in the size and efficiency of steam turbines with 1000 MW now being achieved for a single shaft plant. Figure 2.1 shows the rotor of a modern double-flow low pressure turbine with this power output [8].

Ever, in any case, the gas turbine is generally new. The main commonsense gas turbine used to create power keep running at Neuchatel, Switzerland in 1939, and was produced by the Brown Boveri Company. The main gas turbine fueled plane flight likewise occurred in 1939 in Germany, utilizing the gas turbine created by Hans P. von Ohain. In England, the 1930s' innovation and advancement of the flying machine gas turbine by Frank Whittle brought about a comparable British flight in 1941 [9].

Figure 2.1: The rotor of a modern double-flow low-pressure steam turbine.

The name "gas turbine" is somewhat misleading, because to many it implies a turbine engine that uses gas as its fuel. Actually a gas turbine (as shown schematically in Fig. 2.2 A&B) has a compressor to draw in and compress gas (most usually air); a combustor (or burner) to add fuel to heat the compressed air; and a turbine to extract power from the hot air flow. The gas turbine is an internal combustion (IC) engine employing a continuous combustion process. This differs from the intermittent combustion occurring in Diesel and automotive IC engines. Because the 1939 origin of the gas turbine lies simultaneously in the electric power field and in aviation, there have been a profusion of "other names" for the gas turbine. For electrical power generation and marine applications it is generally called a gas turbine, also combustion turbine (CT), a turboshaft engine, and sometimes a gas turbine engine. For aviation applications it is usually called a jet engine, and various other names depending on the particular engine configuration or application, such as: jet turbine engine; turbojet; turbofan; fanjet; and turboprop or prop jet (if it is used to drive a propeller). The compressor combustor- turbine part of the gas turbine (Fig. 2.2) is commonly termed the gas generator [9].

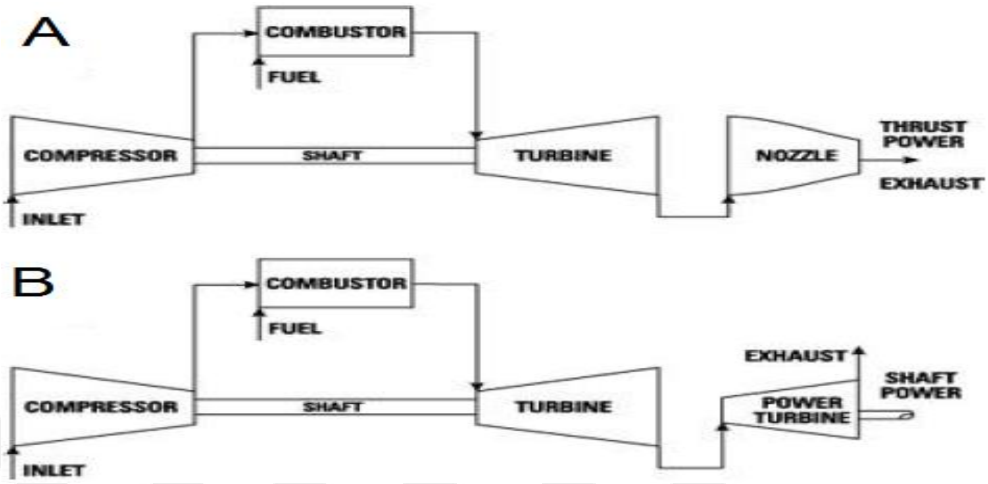

Figure 2.2: Schematic for A) an aircraft jet engine; and B) a land-based gas turbine.

2.2 Definition of a Turbomachine

Turbomachines classified as every one of those gadgets in which vitality is exchanged either to, or from, a persistently streaming liquid by the dynamic activity of one or all the more moving cutting edges lines. The word turbo or turbinis is of Latin birthplace and suggests what twists or spins around. Basically, a pivoting edge push, a rotor or an impeller changes the stagnation enthalpy of the liquid traveling through it by doing either positive or negative work, contingent on the impact required of the machine. These enthalpy changes are personally connected with the weight changes happening all the while in the liquid.

Two principle classes of turbomachine are recognized: to begin with, those that assimilate energy to build the liquid weight or head (ducted and unducted fans, compressor, and pumps); second, those that produce power by extending liquid to lower weight or head (wind, water driven, steam and gas turbine). Figure 2.3 appears, in a straightforward diagrammatic structure, a choice of the numerous assortments of turbomachines experienced practically speaking. The reason that such a variety of various sorts of either pump (compressor) or turbine are being used is a result of the verging on boundless scope of administration prerequisites. For the most part talking, for a given arrangement of working necessities one sort of pump or turbine is the most appropriate to give ideal states of operation.

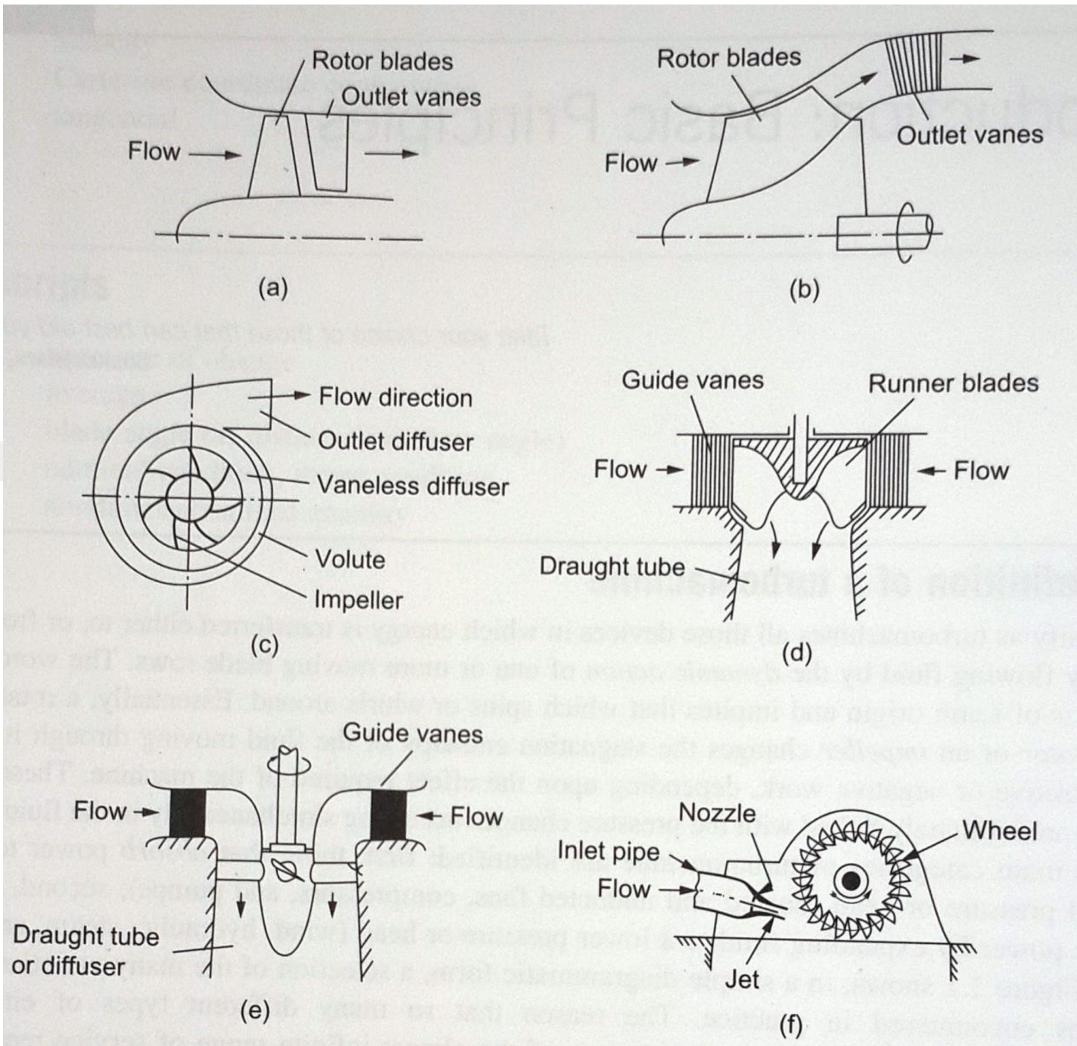

Figure 2.3: Examples of turbomachines.

(a) Single stage axial flow compressor, (b) Mixed flow pump, (c) Centrifugal compressor or pump, (d) Francis turbine (mixed flow type), (e) Kaplan turbine, and (f) Pelton wheel.

Turbomachines are further sorted by nature of the stream way through the sections of the rotor. At the point when the way of the through-stream is entirely or fundamentally parallel to the pivot of revolution, the gadget is named a hub stream turbomachine (e.g., Fig. 2.3 (a) and (e)). At the point when the way or the through-stream is entirely or for the most part in a plane opposite to the pivot hub, the gadget is named a spiral stream turbomachine (e.g., Fig. 2.3 (c)). Blended stream turbomachines are broadly utilized, the term of blended stream alludes to the heading of the trough-stream at the rotor outlet when both outspread and hub speed parts are available in critical sums. Fig. 2.3 (b) demonstrates a blended stream pump and Fig.2.3 (d) a blended stream water powered turbine. All turbomachines can be named either motivation or response machines as per whether weight changes are truant or present, separately, in the move through the rotor. In a motivation machine all the weight change happen in one or more nozzles, the liquid being coordinated onto the rotor. The Pelton wheel, Fig. 2.3 (f), is a case of a drive turbine [8].

2.3 Applications of Turbomachines

Turbomachines are generally utilized as a part of force creating and liquid taking care of frameworks. In a run of the mill focal force plant, fossil or atomic, the focal part is steam turbine, which is utilized to change over the warm vitality of the steam into mechanical vitality to drive an electric generator. A few sorts of pumps are utilized to handle fluid water, including evaporator sustain pump, condensate pump, and cooling-water circling pump. Turbomachines are additionally utilized in other vitality delivering frameworks, for example, hydropower, wind power, and geothermal force establishments.

The other significant use of turbomachines is in the gas turbine motors utilized as a part of air ship and modern force plants. Multistage hub stream gas turbines and compressors are only utilized as a part of high-power units [10].

2.4 Gas Turbine Power Plant

The gas turbine obtains its power by utilizing the energy of burnt gases and air, which is at high temperature and pressure by expanding through the several ring of fixed and moving blades. To get a high pressure (of the request of 4 to 10 bar) of working liquid, which is crucial for development a compressor, is required. The amount of the working liquid and velocity required are all the more, along these lines, by and large, a divergent or a hub compressor is utilized. The turbine drives the compressor thus it is coupled to the turbine shaft. On the off chance that after pressure the working liquid were to be extended in a turbine, then expecting that there were no misfortunes in either segment the influence created by the turbine would be simply equivalent to that consumed by the compressor and the work done would be zero. Yet, expanding the volume of the working liquid at consistent weight, or on the other hand expanding the weight at steady volume can build the force created by the turbine. Including heat so that the temperature of the working liquid is expanded after the pressure may do both of these. To get a higher temperature of the working liquid an ignition chamber is required where burning of air and fuel happens giving temperature ascend to the working liquid. Along these lines, a straightforward gas turbine cycle comprise of

(1) A compressor,

(2) A combustion chamber,

(3) A turbine.

Since the compressor is combined with the turbine shaft, it assimilates a portion of the force created by the turbine and subsequently brings down the proficiency. The system is subsequently the distinction between the turbine work and work required by the compressor to drive it. Gas turbines have been developed to take a shot at the accompanying: oil, characteristic gas, coal gas, maker gas, impact heater and pummeled coal [11].

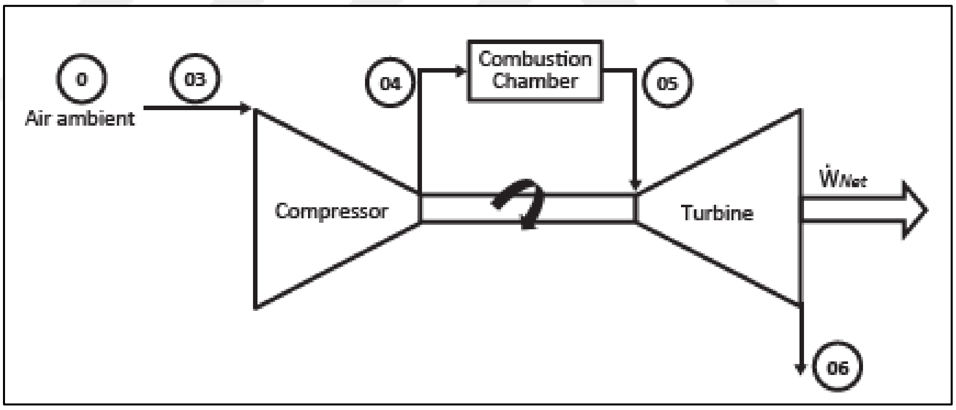

Figure 2.4: Schematic of the typical gas turbine cycle.

The two main factors which affect the performance of gas turbines are the efficiencies of various components and turbine working temperature. The higher they are made, the better is the all-around performance of the plant. It was, in fact low efficiencies and poor turbine materials which caused the failure of a number of early attempts. For example, in 1904 two French engineers built a unit which did little more than turn itself over, with compressor efficiency of about 60%and the maximum gas temperature of about 740 K. The overall efficiency of the gas turbine cycle mainly depends upon the pressure ratio of the compressor. The development of science of aerodynamic and that of metallurgy made it possible to employ very high pressure ratio (20:1) with an adequate compressor efficiency (85-90%) and high turbine inlet temperatures, up to 1500 K [12].

Table 1-1 gives a monetary examination of different era advances from the underlying expense of such frameworks to the working expenses of these frameworks. Since circulated era is extremely site particular the expense will change and the defense of establishment of these sorts of frameworks will likewise differ. Locales for conveyed era shift from huge metropolitan territories to the inclines of the Himalayan mountain range. The financial matters of force era rely on upon the fuel cost, running efficiencies, upkeep expense, and first cost, in a specific order. Site determination relies on upon ecological concerns, for example, emanations, and clamor, fuel accessibility, and size and weight [13].

Table 2.1: Economic comparison of various power generation technologies.

Technology Comparison	Diesel Engine	Gas Engine	Simple Cycle Gas Turbine	Micro Turbine	Fuel Cell	Solar Energy Photovoltic Cell	Wind	Bio-Mass	River Hydro
Product Rollout	Available	Available	Available	Available	1996–2010	Available	Available	2020	Available
Size Range (kW)	20–25,000+	50–7000+	500–450,000+	30–200	50–1000+	1+	10–2500	NA	20–1000+
Efficiency (%)	36–43%	28–42%	21–45%	25–30%	35–54%	NA	45–55%	25–35%	60–70%
Gen Set Cost ($/kW)	125–300	250–600	300–600	350–800	1,500–3,000	NA	NA	NA	NA
Turnkey Cost No-Heat Recovery ($/kW)	200–500	600–1000	300–650	475–900	1,500–3,000	5,000–10,000	700–1300	800–1500	750–1200
Heat Recovery Added Cost ($/kW)	75–100	75–100	150–300	100–250	1,900–3,500	NA	NA	150–300	NA
O & M Cost ($/kWh)	0.007–0.015	0.005–0.012	0.003–0.008	0.006–0.010	0.005–0.010	0.001–0.004	0.007–0.012	0.006–0.011	0.005–0.010

The gas turbine to date in the consolidated cycle mode is quick supplanting the steam turbine as the base burden supplier of electrical force all through the world. This is even valid in Europe and the United States where the expansive steam turbines were the main sort of base burden power in the fossil vitality segment. The gas turbine from the 1960s to the late 1980s was utilized just as cresting force as a part of those nations. It was utilized as base load for the most part as a part of the "creating nations" where the requirement for force was expanding quickly so that the hold up of three to six years for a steam plant was inadmissible [13].

2.5 Gas Turbine Design Considerations

The gas turbine is the most appropriate prime mover when the current necessities, for example, capital cost, time from wanting to finishing, support expenses, and fuel expenses are considered. The gas turbine has the least support and capital expense of any real prime mover. It likewise has the speediest fruition time to full operation of any plant. Its impediment was its high warmth rate however this has been tended to and the new turbines are among the most productive sorts of prime movers. The mix of plant cycles further builds the efficiencies to the low 60s. The configuration of any gas turbine must meet fundamental criteria in light of operational considerations [14]. Main among these criteria are:

1. High efficiency.
2. High reliability and thus high availability.
3. Ease of service.
4. Ease of installation and commission.
5. Conformance with environmental standards.
6. Incorporation of auxiliary and control systems, which have a high degree of reliability.
7. Flexibility to meet various service and fuel needs.

2.6 Categories of Gas Turbines

The simple-cycle gas turbine is classified into five broad groups:

1. Outline Type Heavy-Duty Gas Turbines. The edge units are the huge force era units running from 3 MW to 480 MW in a basic cycle arrangement, with efficiencies extending from 30–46%.

2. Aircraft Derivative Gas Turbines Aero-subordinate. As the name demonstrates, these are force era units, which began in the aeronautic trade as the prime mover of flying machine. These units have been adjusted to the electrical era industry by expelling the detour fans, and including a force turbine at their fumes.

These units range in force from 2.5 MW to around 50 MW. The efficiencies of these units can go from 35–45%.

3. Modern Type-Gas Turbines. These differ in reach from around 2.5 MW–15 MW. This kind of turbine is utilized broadly as a part of numerous petrochemical plants for compressor drive trains. The efficiencies of these units are in the low 30s.

4. Small Gas Turbines. These gas turbines are in the extent from around 0.5 MW–2.5 MW. They frequently have outward compressors and outspread inflow turbines. Efficiencies in the straightforward cycle applications fluctuate from 15–25%.

5. Smaller scale Turbines. These turbines are in the reach from 20 kW–350 kW. The development of these turbines has been sensational from the late 1990s, as there is an upsurge in the disseminated market.

2.7 Gas Turbine Components

It is always necessary for the engineers and designers to know about construction and operation of the components of the gas turbine [11].

2.7.1 Major Gas Turbine Components

2.7.1.1 Compressor

A compressor is a gadget, which pressurizes a working liquid. The sorts of compressors fall into three classifications as appeared in Figure 2.5. The positive relocation compressors are utilized for low stream and high weight (head), radial compressors are medium stream and medium head, and pivotal stream compressors are high stream and low weight. In gas turbines the radial stream and pivotal stream compressors, which are nonstop stream compressors, are the ones utilized for compacting the air. Positive removal compressors, for example, the apparatus sort units are utilized for oil frameworks as a part of the gas turbines. Compressor proficiency is imperative in the general execution of the gas turbine as it devours 55–60%of the force produced by the gas turbine [13]. A multistage axial-flow compressor is extensively used in gas turbine engines, especially for aircraft propulsion compared with centrifugal type [10].

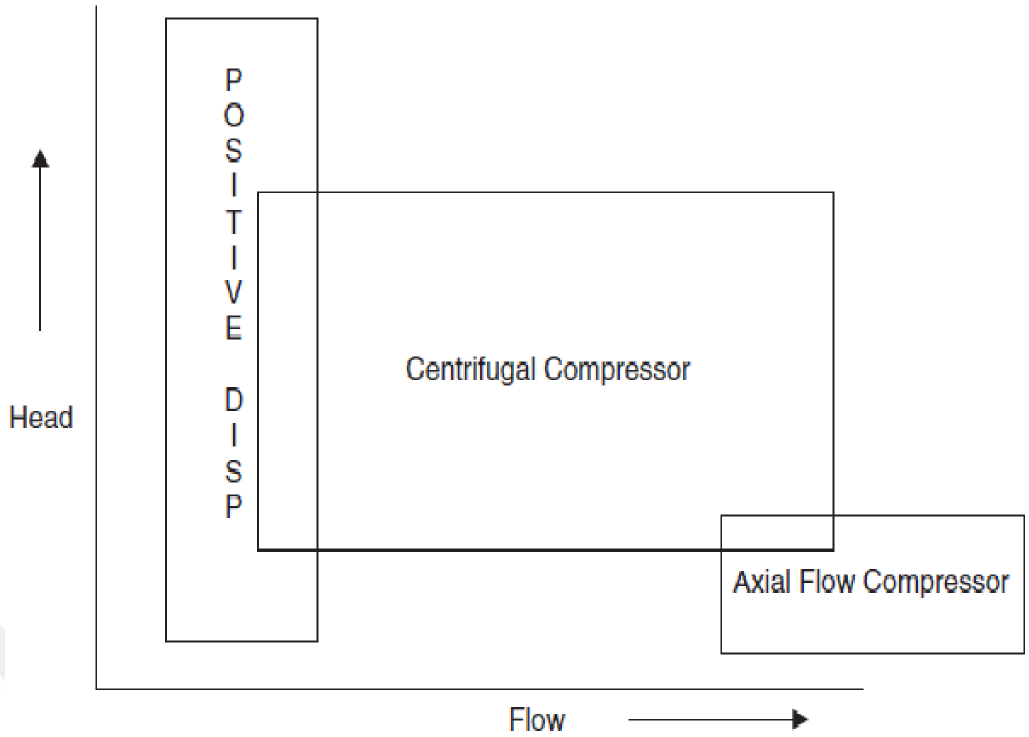

Figure 2.5: Performance characteristics of different types of compressor.

2.7.1.2 Combustion chamber (combustor)

All gas turbine combustors perform the same function: they increase the temperature of the high-pressure gas. Combustion is the chemical mixture of a substance with definite factors, frequently oxygen, accompanied by a high temperature production or heat transfer. The combustion chamber task is to receive the air from the compressor and to supply it to the turbine at the involved temperature, perfectly with no waste of pressure. Fundamentally, it is a direct-fired air heater in which fuel is flamed with less than one-third of the air after which the combustion products are then mixed with the remaining air. For the public open-cycle gas turbine, this wants the inner combustion of fuel. This means the problem of fuel operation, mixing and burning, must be addressed. Fuel is usually liquid or gaseous. Fuels liquid or gaseous are frequently hydrocarbons. Liquids may range from refined gasoline during kerosene and light diesel oil to a heavyweight remaining oil. Gases frequently being generally methane, natural gas, and butane. Combustion has a huge difficulties. The difficulty appears due to loss of pressure in combustion chamber. Nearly any fuel can be flamed effectively if enough pressure drop is available to supply the required turbulence for air and fuel mixing and if adequate volume is available to give the basic time for combustion to be achieved.

The gas turbine is a constant stream framework; consequently, the burning in the gas turbine contrasts from the ignition in diesel motors. High rate of mass stream results in high speeds at different focuses all through the cycle (300 m/sec). One of the indispensable issues connected with the configuration of gas turbine burning framework is to secure a consistent and stable fire inside the ignition chamber. The gas turbine ignition framework needs to work under certain diverse working conditions which are not for the most part met with the burning frameworks of diesel motors. A couple of them are recorded underneath:-

1. Burning in the gas turbine happens in a nonstop stream framework and, in this manner, the benefit of high weight and limited volume accessible in diesel motor is lost. The synthetic response happens moderately gradually in this way requiring vast habitation time in the burning chamber with a specific end goal to accomplish complete ignition.

2. The gas turbine requires around 100:1 air-fuel proportion by weight for the reasons said before. Be that as it may, the air-fuel proportion required for the burning in diesel motor is around 15:1. In this way, it is difficult to touch off and keep up a persistent ignition with such frail blend. It is important to give rich blend of ignition and persistent burning, and along these lines, it is important to permit required air in the burning zone and the rest of the air must be added after complete ignition to diminish the gas temperature before going into the turbine.

3. A pilot or recalculated zone ought to be made in the primary stream to set up a steady fire that touches off the burnable blend consistently.

4. A stable ceaseless fire can be kept up inside the ignition chamber when the stream speed and fuel smoldering speed are equivalent. Sadly the majority of the fills have low blazing speeds of the request of a couple meters for every second, in this manner, fire adjustment is impractical unless some method is utilized to stay the fire in the ignition chamber.

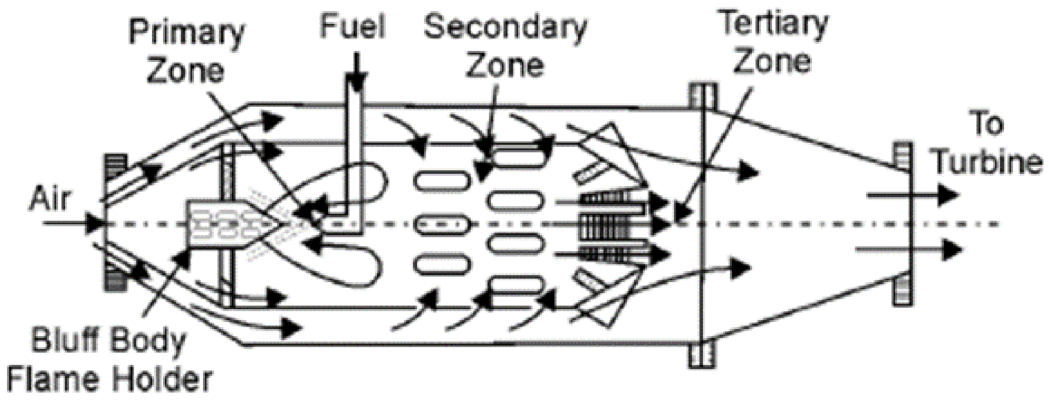

Figure 2.6: Combustion chamber with upstream injection with bluff-body flame holder.

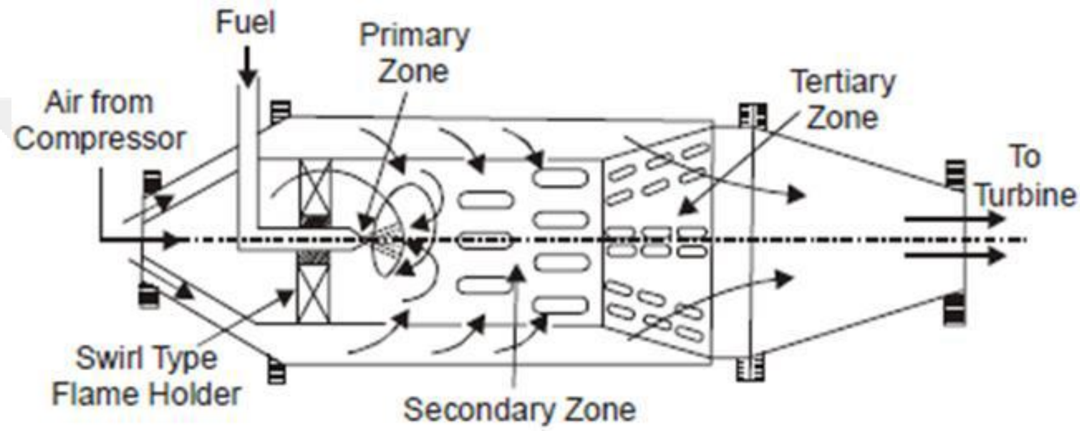

Figure 2.7: Combustion chamber with downstream injection and swirl holder.

The common methods of flame stabilization used in practice are bluff body method and swirl flow method. Two types of combustion chambers using bluff body and swirl for flame stabilization are shown in Figure 2.6 and Figure 2.7. The significant contrast between two is the utilization of various techniques to make pilot zone for fire adjustment. About 15 to 20%of the aggregate air is passed around the plane of fuel giving rich blend in the essential zone. This blend smolders persistently in the essential (pilot) zone and delivers high temperature gasses. Around 30%of the aggregate air is supplied in the auxiliary zone through the annuals around the fire tube to finish the ignition. The optional air must be conceded at right focuses in the ignition chamber generally the icy infused air may cool the fire locally along these lines diminishing the rate of response. The auxiliary air finishes the ignition and also cools the fire tube. The staying half air is blended with smoldered gasses in the "tertiary zone" to chill the gasses off to the temperature suited to the turbine cutting edge materials. By inserting a bluff body in standard, a low-weight zone is made downstream side that causes the inversion of stream along the pivot of the ignition chamber to balance out the fire. If there should be an occurrence of twirl adjustment, the essential air is gone through the whirled, which delivers a vortex movement making a low-weight zone along the pivot of the chamber to bring about the inversion of stream. Adequate turbulence must be made in each of the three zones of burning and uniform blending of hot and frosty bases to give uniform temperature gas stream at the outlet of the ignition chamber [11].

2.7.1.3 Turbine

Turbine is a kind of converter, through the rotor, kinetic energy and some thermal energy are transferred to mechanical energy to drive the compressor, fan and other accessories. For power gas turbine, all useful output power is the excessive shaft power of the turbine rotor available to drive the external machine [10]. There are two sorts of turbines utilized as a part of gas turbines. These comprise of the pivotal stream sort and the outspread inflow sort. The hub stream turbine is utilized as a part of more than 95%of all applications. The two sorts of turbine pivotal stream and spiral inflow turbines can be isolated further into drive or response sort units. Drive turbines take their whole enthalpy drop through the nozzles, while the response turbine takes a halfway drop through both the nozzles and the impeller blades [14].

The fundamental necessities of the turbines are lightweight, high productivity; dependability in operation and long working life. Extensive work yield can be gotten per stage with high cutting edge speeds when the sharp edges are intended to support higher burdens. More phases of the turbine are constantly favored in gas turbine power plant since it diminishes the anxieties in the cutting edges and expands the general existence of the turbine. More stages are further favored with stationary force plants since weight is not the significant thought in the configuration which is fundamental in flying machine turbine-plant [11].

2.7.2 Minor Gas Turbine Components

Auxiliary systems are the foundation of the gas turbine plant. Without assistant framework, the very presence of the gas turbine is unthinkable. It allows the sheltered working of the gas turbine. The assistant framework incorporates beginning, ignition, oil and fuel framework and control.

2.7.2.1 Starting systems

Two separate frameworks beginning and ignition are required to guarantee a gas turbine motor will begin palatably. Amid motor beginning the two frameworks must work at the same time.

(*a*) Electrical

A.C. cranking motors are usually 3 phase induction types rated to operate on the available voltage and frequency.

D.C. starter motor takes the source of electrical energy from a bank of batteries of sufficient capacity to handle the starting load. Engaging or disengaging clutch is used.

(*b*) Pneumatic or Air Starter

Air beginning is utilized for the most part as it is light, basic and temperate to work. As air starter engine has a turbine rotor that transmits power through a decrease apparatus and grip to the starter yield shaft that is associated with the motor.

(*c*) Combustion Starter

It is in each appreciation a little gas turbine. It is a totally coordinated framework which joins a planetary lessening gear drive with over-running grip. The unit is begun with the electric starter. The starter turbine is specifically equipped to the gas turbine shaft through a lessening gear.

(*d*) Hydraulic Starting Motor

It consists of a hydraulic starter motor for main engine, an accumulator, a hydraulic pump motor for auxiliary power unit (A.P.U.).

2.7.2.2 Ignition system

Ignition system is used to start flash amid the beginning. When it begins, the burning is ceaseless and the working of ignition framework is cut-off naturally.

2.7.2.3 Lubrication system

The sifted oil streams to the course and rigging instance of turbine and subsequent to greasing up and cooling the oil washed parts, it is come back to the oil tank through a cooler by a rummage pump.

2.7.2.4 Control system

The purpose of gas turbine controls is to meet the specific control requirements of users and safe operation of the turbine. There are basically two types of controls. They are as follows.

2.7.2.4.1 Prime control

The target of the prime control is to guarantee the best possible utilization of the turbine energy to the heap. The clients of the gas turbines have particular control necessities concurring the utilization of gas turbines. The necessities may be to control:

(1) The frequency of an A.C. generator,

(2) The speed of a boat or ship,

(3) The speed of an aircraft,

(4) The capacity or head of a pump or compressor,

(5) The road speed of a vehicle.

2.7.2.4.2 Protective control

The target of the defensive control is to guarantee sufficient security for the turbine in keeping its operation under antagonistic conditions. At whatever point, perilous working conditions are drawn nearer, the prime control is overwhelmed by the defensive control to secure the turbine or driven hardware.

2.8 Theory of Operation

The Brayton cycle was initially proposed by George Brayton for use in the responding oil-copying motor that he created around 1870. Today, it is utilized for gas turbines just where both the pressure and extension forms occur in pivoting apparatus. Gas turbines typically work on an open cycle, as appeared in Fig. 2.8. Outside air at encompassing conditions is drawn into the compressor, where its temperature and weight are raised. The high-weight air continues into the ignition chamber, where the fuel is blazed at consistent weight. The subsequent high-temperature gasses then enter the turbine, where they extend to the barometrical weight while creating power. The fumes gasses leaving the turbine are tossed out (not recycles), bringing on the cycle to be delegated an open cycle. The open gas-turbine cycle depicted above can be demonstrated as a shut cycle, as appeared in Fig.2.9, by using the air-standard suspicions. Here the pressure and development forms continue as before, yet the burning procedure is supplanted by a steady weight heat-expansion process from an outer source, and the fumes procedure is supplanted by a consistent weight heat-dismissal procedure to the encompassing air.

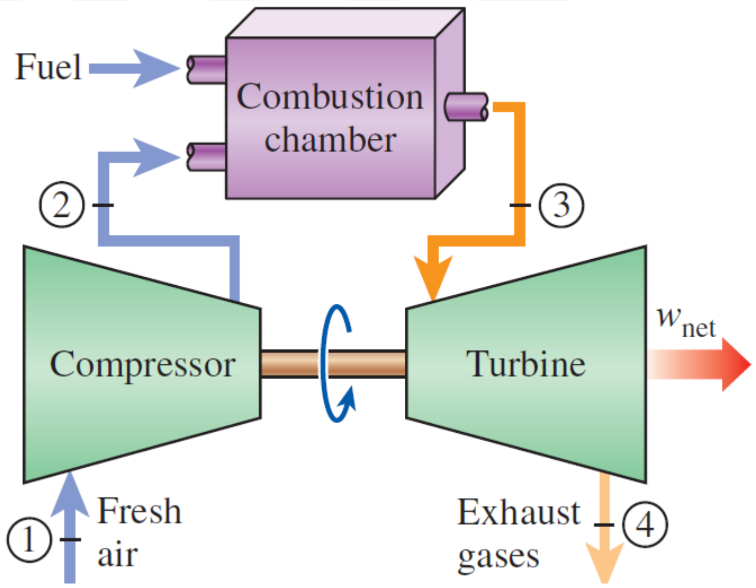

Figure 2.8: An Open-cycle Gas-turbine engine.

The ideal cycle that the working fluid undergoes in this closed loop is the Brayton cycle, which is made up of four internally reversible processes:

1-2 Isentropic compression (in a compressor)

2-3 Constant-pressure heat addition

3-4 Isentropic expansion (in a turbine)

4-1 Constant-pressure heat rejection.

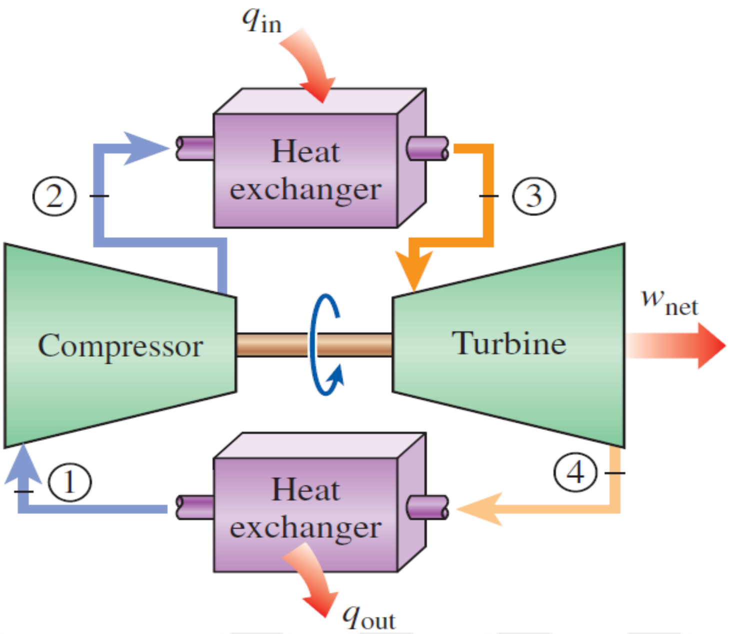

Figure 2.9: A closed-cycle Gas-turbine engine.

The *T-s* and *P-v* diagrams of an ideal Brayton cycle are shown in Fig. 2.10. Notice that all four processes of the Brayton cycle are executed in steady-flow devices; thus, they should be analyzed as steady-flow processes [15].

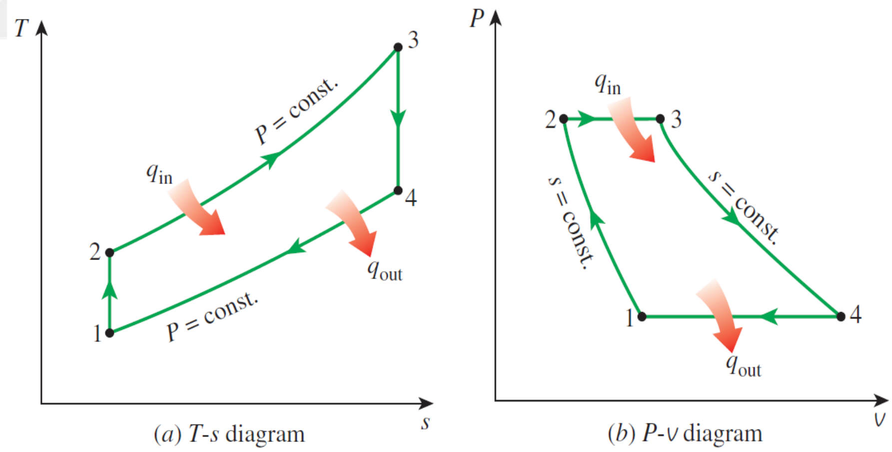

Figure 2.10: T-s and P-*v* diagram for the ideal Brayton cycle.

2.9 Deviation of Actual Gas-Turbine Cycles from Idealized Ones

The genuine gas-turbine cycle contrasts from the perfect Brayton cycle on a few records. First off, some weight drop amid the warmth expansion and warmth dismissal procedures is inescapable. All the more vitally, the genuine work contribution to the compressor is more, and the real work yield from the turbine is less in light of irreversibilities. The deviation of genuine compressor and turbine conduct from the glorified isentropic conduct can be precisely represented by using the isentropic efficiencies of the turbine and compressor as.

$$\eta_C = \frac{w_s}{w_a} \cong \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

$$\eta_T = \frac{w_a}{w_s} \cong \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

where states 2*a* and 4*a* are the actual exit states of the compressor and the turbine, respectively, and 2*s* and 4*s* are the corresponding states for the isentropic case, as illustrated in Fig. 2.11.

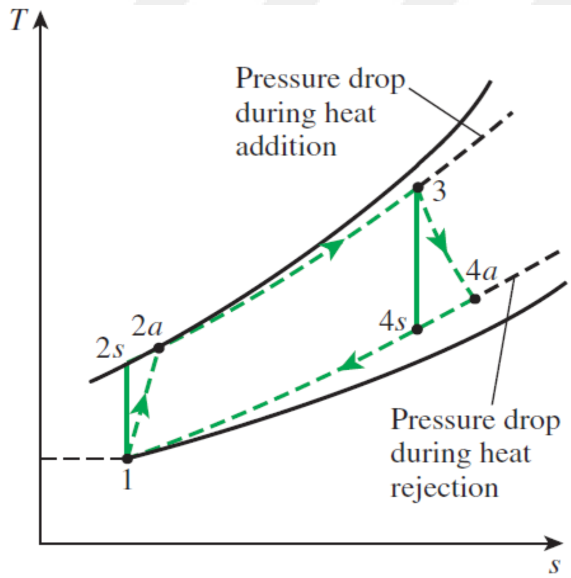

Figure 2.11: The deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities.

2.10 Factors Affecting Gas Turbine Performance

We know the operations of our gas turbines wouldn't all be able to be flawless, and we'll gone through estimations, practicality studies and more to pinpoint the careful cause. In any case, before the majority of that is proficient, you ought to keep a rundown in the back of your psyche of what may bring about your misfortune in execution, in light of regular variables that influence gas turbine productivity and that's just the beginning.

2.10.1 Air Temperature and Site Elevation

Since a gas turbine is an air-breathing machine, its execution is changed by anything that influences the thickness and/or mass stream of the air admission to the compressor. Encompassing climate conditions are the most clear transforms from the weight and temperature of reference, individually 14.7psi/1.013bar and 59F/15C [16].

2.10.2 Humidity

Since humid air is less thick than dry air this straightforwardly influences the execution of gas turbines. Accordingly, for the same volume entering the gas turbine, diminished mass stream rate will be utilized, decreasing the force yield.

2.10.3 Inlet and Exhaust Losses

Embedding air filtration, hushing, evaporative coolers or chillers into the delta or warmth recuperation gadgets in the fumes causes weight misfortunes in the framework.

2.10.4 Fuels

Work from a gas turbine can be characterized as the result of mass stream, heat vitality in the combusted gas (Cp) and temperature differential over the turbine. The mass stream in this condition is the aggregate of compressor wind current and the infused fuel stream. The warmth vitality is a component of the components in the fuel and the results of ignition. Accordingly distinctive fills will hold diverse force.

2.10.5 Fuel Heating

Fuel warming will bring about a somewhat bring down gas turbine yield in light of the incremental volume stream diminish. Since utilization of this vitality in the gas turbine fuel warming framework is thermodynamically profitable, the consolidated cycle proficiency can be moved forward.

2.10.6 Air Extraction

In a few gas turbine applications, it might be attractive to concentrate air from the compressor (cooling for the turbine, pressurized air for the lodge in a plane, and so on.) Generally, up to 5% of the compressor wind stream can be separated from the compressor release packaging without adjustment to housings or on-base funneling. Be that as it may, as the air extraction expands, the force created actually diminishes because of the restricted mass stream rate experiencing the turbine.

CHAPTER THREE

DUHOK GAS POWER STATION

3.1 Introduction

Duhok governorate one of the northern provinces in Iraq, and the third city in the province of Kurdistan Region. It is far away 488 Km from the capital Baghdad, rises 500 m above the sea level.

3.2 Duhok Gas Power Station (DGPS)

DGPS is located to the north of Duhok governorate near Sumel city, on Duhok-Zakho highway around 35 km from Duhok city center. Owned, Build and Operate by the MASS GROUP HOLDING. The plant consists of eight Frame 9E gas turbines with a rated capacity of 125MW each and 1000MW in total. In the beginning of 2010, the Ministry of Electricity (MoE) of the Kurdistan Region contracted MGH to build the Duhok Gas Power Station (DGPS) on the basis of Build, Own and Operate (BOO). Accordingly, MGH purchased units of Frame-9E turbines from General Electric Company (GE) with their generators and auxiliaries. The first unit of phase one of the project connected to the national grid in Kurdistan region on December 18, 2010 and the last unit (number eight) connected to the grid in 14th August 2013. MGH contracted ABB Germany as the main EPC contractor, for installation, testing and commissioning of the plant. ABB Company supplied the Balance Of Plant (BOP) equipment for the project.

ABB Sweden executed the supply, installation, testing and commissioning of the 132 KV Voltage Transfer Station in DGPS. Similar to the ones in Erbil and Suleimaniah, DGPS is designed to run on two types of fuel, natural gas as main fuel, and diesel fuel as standby.

Since the start of operating to date the plant is running on diesel delivered by trucks, stored in tanks via 10 unloading pumps inside the plant in a storage areas prepared for this purpose. Then the diesel is treated and purified prior to its use in the turbines [17].

Figure 3.1: Duhok gas power station.

3.3 Gas Turbine unit in DGPS

The MS9001E gas turbine is GE's 50 Hz workhorse. With more than 430 units, it has accumulated over 18 million fired hours of utility and industrial service, many in arduous climates ranging from desert heat and tropical humidity to arctic cold. Originally introduced in 1978 at 105 MW, the 9E has incorporated numerous component improvements. The latest model boasts an output of 126 MW and is capable of achieving more than 52%efficiency in combined cycle.

Whether for simple cycle or combined cycle application, base load or peaking duty, 9E packages are comprehensively engineered with integrated systems that include controls, auxiliaries, ducts and silencing.

They are designed for reliable operation and minimal maintenance at a competitively low installed cost.

Like GE's other E-class technology units, the Dry Low NOx combustion system is available on 9E, which can achieve NOx emissions under 15 ppm when burning natural gas. With its flexible fuel handling capabilities, the 9E accommodates a wide range of fuels, including natural gas, light and heavy distillate oil, naphtha, crude oil and residual oil. Designed for dual-fuel operation, it is able to switch from one fuel to another while running under load. It is also able to burn a variety of syngas produced from oil or coal without turbine modification. This flexibility, along with its extensive experience and reliability record, makes the 9E well suited for IGCC projects.

Fig. 3.2, shows the MS9001E which is a reliable, low first-cost machine for peaking service, while its high combined cycle efficiency gives excellent fuel savings in base load operations. Its compact design provides flexibility in plant layout as well as the easy addition of increments of power when a phased capacity expansion is required.

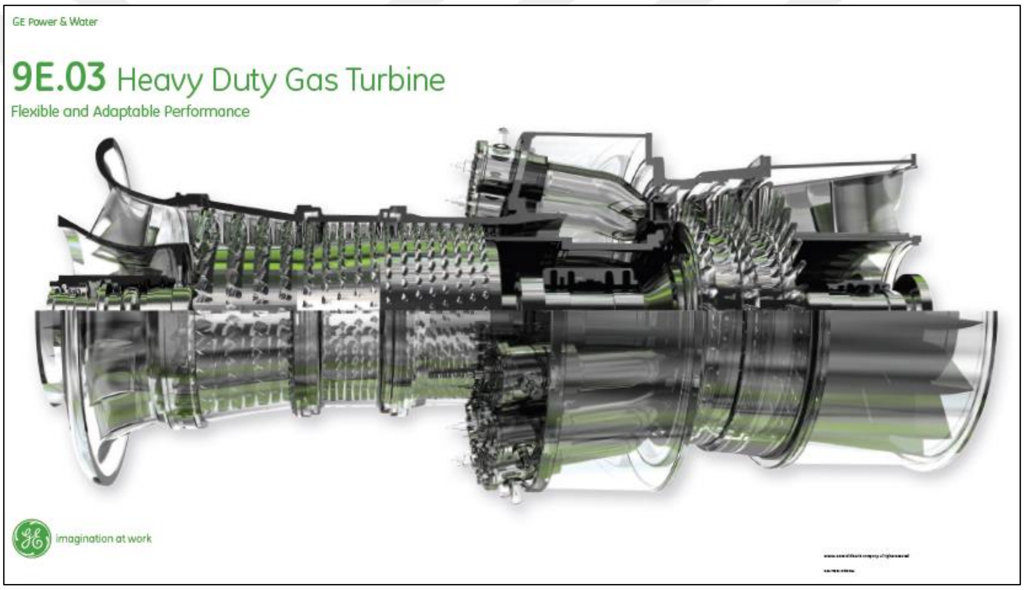

Figure 3.2: GE 9E.03 heavy duty gas turbine.

Fig. 3.3, illustrate GE's portfolio of heavy duty and aeroderivative gas turbines helps provide a sense of certainty in an uncertain world, delivering operational flexibility and performance needed to adapt to a rapidly evolving power generation environment. With gas turbine products ranging in individual output from 22 MW to 519 MW, GE has a solution to reliably and efficiently deliver the power needed by utility power generators, industrial operators, and communities. Even in remote locations and harsh conditions, you can count on GE to deliver a gas turbine that will meet your needs.

All gas turbines share the common heritage of jet engine technology pioneered by GE in the first half of the 20th century. They are typically categorized as either heavy duty (sometimes also called “frame”) or aeroderivative gas turbines, although some turbines recently have adopted features of both design types. In general, the differences between the aeroderivative and heavy duty gas turbines are weight, size, combustor type, and turbine design. Heavy duty gas turbines are usually field constructed and maintained in place, whereas aeroderivative gas turbines are designed to allow for quick replacement of the entire engine when maintenance is required [18].

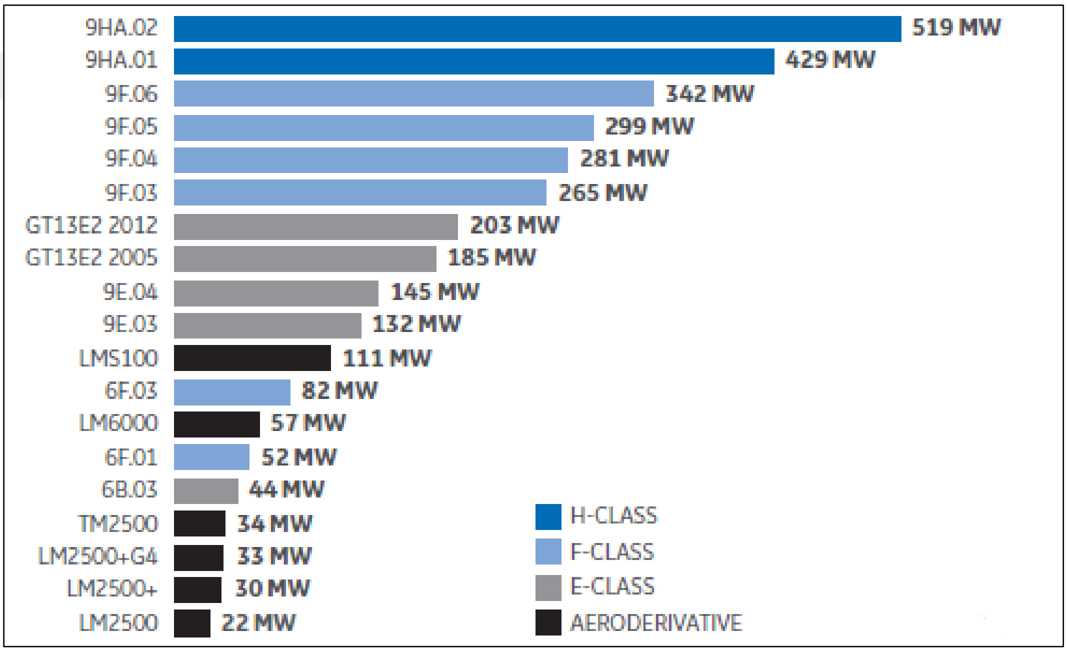

Figure 3.3: GE's 50 Hz portfolio of heavy duty and aeroderivative gas turbine.

3.4 Gas Turbine GE 9E.03 Technical Design Data

High accessibility and dependability are the most imperative parameters in the outline of a gas turbine. The accessibility of a force plant is percent of time the plant is accessible to create power in any given period. The unwavering quality of the plant is the rate of time between arranged redesigns [13].

Table 3.1 listed below illustrate the technical design data for GE 9E.03 used in DGPS.

Table 3.1: GT GE9E.03 manufacturer technical design data.

GT GE 9E.03 Technical Design Data			
		Units	Value
SC plant performance	Net Output	MW	132
	Net Heat Rate	Btu/kwh, LHV	9,860
	Net Heat Rate	KJ/ kwh, LHV	10,403
	Net Efficiency	%	34.6
	Pressure Ration		12.6
	Turbine Speed	rpm	3000
Gas Turbine Performance	Exhaust Temperature	°F	1,012
	Exhaust Temperature	°C	544
	Exhaust Energy	MM Btu/hr.	828
	Exhaust Energy	MM kJ/hr.	847
	GT Turndown Min. Load	%	35
	GT Ramp Rate	MW/min.	50
	CO at min. Turndown	ppm	25
	Wobbe Variation	%	>+/-30
	Startup time	Conventional/Peaking, Minutes	30/10
1x CC Plant Performance	Net Output	MW	201
	Net Heat Rate	Btu/kwh, LHV	6,460
	Net Heat Rate	kJ/kwh, LHV	6,816
	Net Efficiency	%	52.8
	Plant Turndown	%	46
	Ramp Rate	MW/min	50
	Startup Time	Minutes	38
2x CC Plant Performance	Net Output	MW	405
	Net Heat Rate	Btu/kwh, LHV	6,410
	Net Heat Rate	kJ/kwh, LHV	6,763
	Net Efficiency	%	53.2
	Plant Turndown	%	22
	Startup Time	Minutes	38

3.5 Operational Data of the Duhok Gas Power Station

Data used for this study were collected from the power plant's log book and final report such as average daily power generated, mass flow rate, pressure and temperature. Working fluid parameters (air and fuel specific capacities and universal constants) were obtained from appropriate thermodynamic tables as shown in the table 3.2 below.

Table 3.2: Sample of the Duhok Gas Power Station data sheet.

DGPS		Duhok Gas Power Station MGH - DGPS - OPS - RPT - 0001								
		GT1								
		Fuel Consumption	Mega Watt Hours	Inlet Air Temp.	Comp. Discharge Temp.	Inlet Air DP	Comp. Discharge Pressure	MEGA WATT	Mega Var	Base Preselect Off
	Time	Ton	(MWH)	(°C)	(°C)	MM H20	(BAR)	MW	(MVAR)	Status
25 MAY 2013	03:00	502604	1753180	27.0	307.0	39.0	7.0	30.0	4.0	PRE
	04:00	502618	1753210					30.0		PRE
	05:00	502631	1753240	26.0	306.0	39.0	7.0	30.0	5.0	PRE
	06:00	502645	1753270					40.0		PRE
	07:00	502661	1753310	27.0	330.0	51.0	8.0	50.0	10.0	PRE
	08:00	502678	1753360					70.0		PRE
	09:00	502701	1753440	29.0	352.0	51.0	9.0	80.0	21.0	PRE
	10:00	502724	1753520					90.0		PRE
	11:00	502751	1753620	31.0	367.0	53.0	10.0	106.0	27.0	BAS
	12:00	502773	1753740					105.0		BAS
	13:00	502811	1753830	32.0	370.0	54.0	10.0	106.0	26.0	BAS
	14:00	502840	1753930					106.0		BAS

CHAPTER FOUR

AIR INLET TEMPERATURE AND COOLING TECHNOLOGY

4.1 Introduction

Industrial gas turbines that operate at constant speed are constant-volume-flow combustion machines. As the specific volume of air is directly proportional to the temperature, the increases of the air density results in a higher air mass flow rate, once the volumetric rate is constant. Inlet air-cooling increases the power output by taking advantage of the gas turbine’s feature of higher mass flow rate, due the compressor inlet temperature decays. Consequently, the gas turbine power output enhances [4].

4.2 The influence of Air Temperature on Gas Turbine Performance

Power output and efficiency of which is highly affected by the environmental conditions. When the inlet temperature rises, the temperature ratio and equivalent flow of air and reduced speed of gas turbine will decrease, which will result in performance of the gas turbine degeneration.

The characteristic that gas turbine power plant performance is affected by the ambient air temperature has aroused widespread attention. If it were possible to obtain a constant low inlet air temperature, a constant high power output could be generated from a gas turbine. Now inlet-cooling technologies has been applied in practice.

4.2.1 Analysis of the Influence of the Temperature on Gas Turbine Performance

Atmospheric temperature has great influences on power output and efficiency of simple cycle of gas-powered plant. At present, the number of importing heavy-duty gas turbines is increasing year by year, therefore it is very significant and valuable to study on the effects of temperature on large power performance. Taking GE PG9371FB, Mitsubishi M701F4, Siemens SGT5-4000F, which are the most advanced heavy-duty gas turbine model. Some aspects of the mentioned issues are being discussed, as shown in Figs. 4.1,2, and 3. As we can see from the Figure 4.2, the output of these three kinds of gas turbine units reduce greatly as the temperature rises. Taking GE PG9371FB as an example, for gas turbine simple cycle, the load at 28 °C is approximately about 30 MW lower than ISO design condition (15 °C), similarly, for gas-steam combined cycle, the load at 28 °C decrease about 40 MW.

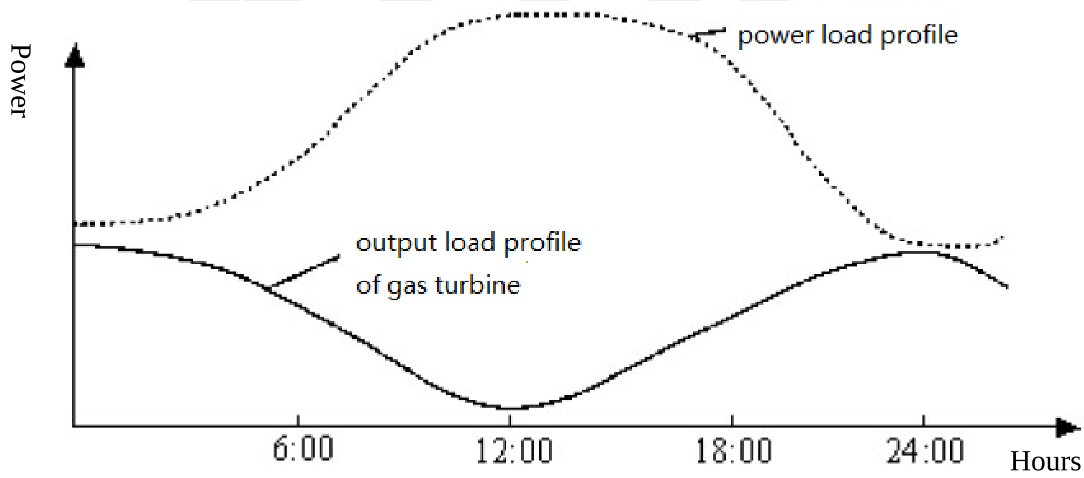

Figure 4.1: Daily Relation curve between power load and power output gas turbine.

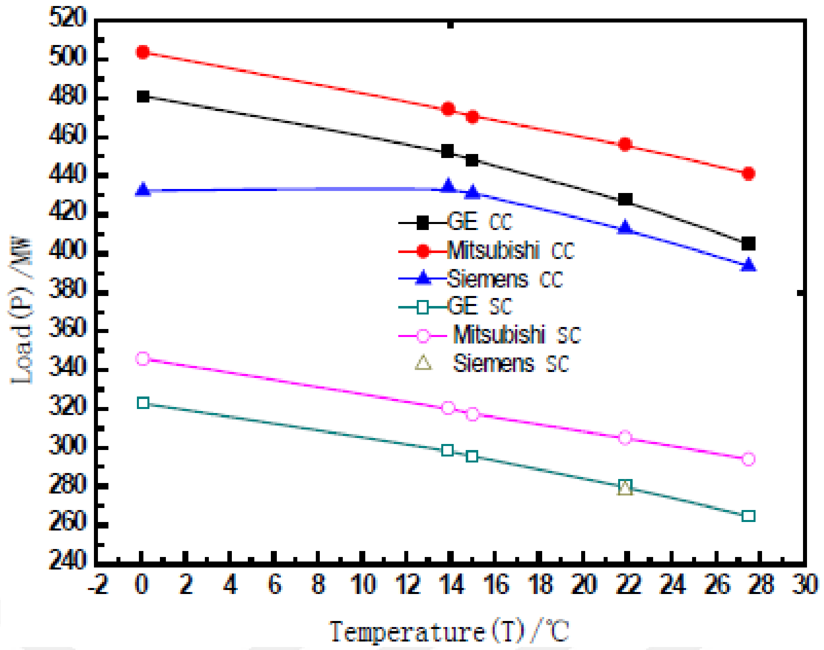

Figure 4.2: P-T curves of GSCC units at full power load.

Figure 4.3 shows that temperature has a greater influence on efficiency of those three kinds of model's combined cycle. The influence on GE unit is most prominent. When the temperature is 28 °C, its efficiency is 57.3 %, which is 1%lower than ISO design condition. From the analysis, we can get a conclusion that temperature has a very significant influence on the performance of gas turbine units.

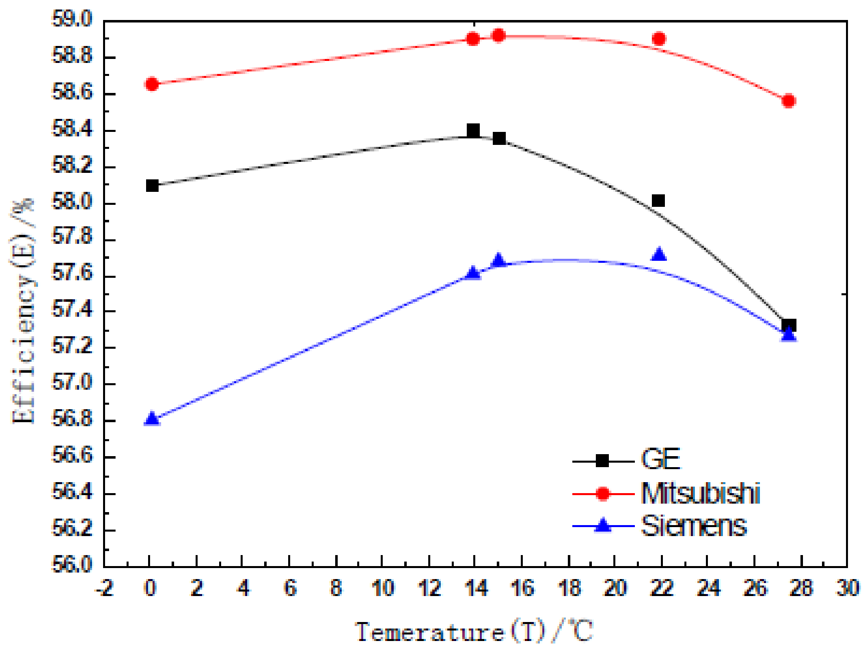

Figure 4.3: Efficiency –Temperature curves of GS units at base load.

4.2.2 Theoretical Analysis of Effecting of the Temperature on Performance of Gas Turbine

As the inlet air temperature rise, efficiency and power output of the gas turbine will reduce. Conversely, they will increase when the temperature drops. For simple cycle gas turbine, the main reasons are as following:

1) When the temperature of the air increase, the air density and, consequently, the air mass flow decreases. The reduced air mass flow directly causes the gas turbine to produce less power output. The ideal gas state equation,

$$\dot{m} = \rho * \dot{v} \qquad \textbf{(1)}$$

$$\rho = \frac{P}{RT} \qquad \textbf{(2)}$$

$$\dot{m} = \frac{P\dot{v}}{RT} \qquad \textbf{(3)}$$

Where $\dot{m}$, ρ, $\dot{v}$, P, R, T are air mass flow rate, density of the air, volumetric flow rate, air pressure, ideal gas constant and air temperature respectively. The air pressure basically remains constant or with negligible changing. As we can see in equation (3), mass of air flow rate is proportional inversely with the air temperature. When the inlet temperature coming down, air density will rise obviously and the air mass will increase, so the power output will also increase accordingly.

2) By the thermodynamic, the decries of the gas temperature is, the lower of its molecular movement. So the smaller compression work is needed to achieve the same pressure. The equations are as follows:

$$Comressor\ Work = \dot{m}_a * C_p * (T_{outlet} - T_{inlet}) \qquad \textbf{(4)}$$

$$W_c = C_p T_1({p_r}^m - 1) \qquad \textbf{(5)}$$

$$W_{GT} = W_T - W_C = C_p T_1[\tau(1 - {p_r}^{-m}) - ({p_r}^m - 1)] \qquad \textbf{(6)}$$

$$\lambda = 1 - \frac{W_C}{W_T} = 1 - \frac{T_1({p_r}^m - 1)}{T_3(1 - {p_r}^{-m})} = 1 - \frac{{p_r}^m}{\tau} \qquad \textbf{(7)}$$

Where, W_{GT} is power output of the ideal cycle, W_T is expansion power, W_C is consumption work of compressor, C_p is air specific heat at constant pressure, T_1 is atmospheric temperature, p_r is pressure ratio, and τ is temperature ratio. *m=(k-1)/k*. *k* is adiabatic index, $\boldsymbol{\lambda}$ is coefficient of useful work, and T_3 turbine inlet Temperature

Because the gas turbine output power is the difference between gas turbine expansion power and compressor consumption work from the above equations, the higher intake-air temperature results in an increase of the specific compressor work and, therefore, in a further reduction of the power output [19].

4.3 Gas Turbine Cooling Systems Technologies

The performance deterioration of gas turbine in terms of power output and thermal efficiency at high ambient air temperatures, correlative with time period of high electricity demand, implies that the integration of an intake air-cooling system could be an important consideration to boost the performance of the plant [3].

Gas turbine cooling technologies can be divided into two categories:

1) Evaporative cooling, which is mainly divided into media type evaporative cooling and the fogging system cooling, based on the structure of the cooler.

2) Refrigeration cooling, which is mainly divided into compression refrigeration cooling, absorption refrigeration cooling and storage according to the ways to obtain source. This thesis focuses on the analysis of media type evaporative cooling and absorption cooling which are typical two kinds of the inlet air cooling technologies [19].

4.4 Media type Evaporative Cooling System

Media type evaporative cooling system for gas turbine inlet air is a useful option for installation where high ambient temperature and low relative humidity are common. With an evaporative cooler, water is added to the inlet air of gas turbine. Most of the water evaporates absorbing latent heat from the air. As a result, air which gives up sensible heat cools and increase in density. This gives the machine a higher mass flow rate and pressure ratio resulting in an increase in turbine output and efficiency [20].

Water used with evaporative coolers often contains dissolved solids such as sodium and potassium, which, in combination with sulfur in the fuel, are principal ingredients in hot gas path corrosion. For this reason, water quality and the prevention of water carry-over are important considerations in the use of evaporative coolers. The prevention of water carry-over is accomplished by correct design of the evaporative cooler, and proper installation and operation. Water quality requirements depend on the amount of water carry-over expected (or allowed) and can vary from the use of deionized water to water with significant concentrations (as much as several hundred ppm by weight, in water) of sodium and potassium [21].

The evaporative water mass flow associated with the evaporative cooling is given by:

$$\dot{m}_w = \dot{m}_a * (\omega_{inlet} - \omega_{outlet}) \quad \textbf{(8)}$$

where $\dot{m}_a$ is mass flow rate of the air, $\omega_{inlet}, \omega_{outlet}$ are the air specific humidity in the inlet and outlet of the evaporative system, respectively. The water that drained from cooling equipment to remove mineral build-up is called "bleed" water. Eq. (8) not consider the bleed water [7].

4.4.1 Theory of Operation for the Evaporative Cooling System

Evaporative cooling involves heat and mass transfer, which occurs when water and the unsaturated air water mixture of the incoming air are in contact. This transfer is a function of the differences in temperatures and vapor pressures between the air and water. Heat and mass transfer are both operative in the evaporative cooler because heat transfer from the air to the water evaporates water, and the water evaporating into the air constitutes mass transfer. Heat inflow can be described as either latent or sensible heat. Whichever term is used depends on the effect. If the effect is only to raise or lower temperature, it is sensible heat. Latent heat, on the other hand, produces a change of state, e.g., freezing, melting, condensing, or vaporizing. In evaporative cooling, sensible heat from the air is transferred to the water, becoming latent heat as the water evaporates. The water vapor becomes part of the air and carries the latent heat with it. The air dry-bulb temperature is decreased because it gives up sensible heat. The air wet-bulb temperature is not affected by the absorption of latent heat in the water vapor because the water vapor enters the air at the air wet-bulb temperature. Theoretically, the incoming air and the water in the evaporative cooler may be considered an isolated system. Because no heat is added to or removed from the system, the process of exchanging the sensible heat of the air for latent heat of evaporation from the water is adiabatic. Evaporative cooler performance is based on the concept of an adiabatic process [7].

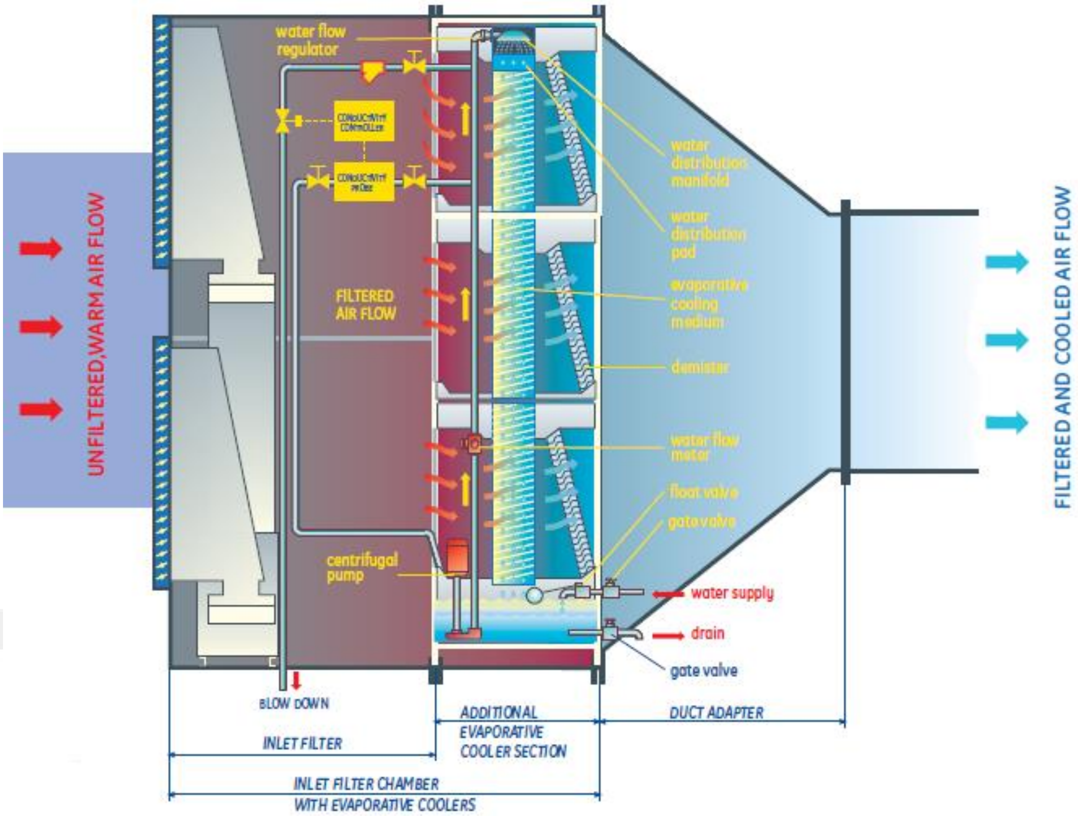

Figure 4.4: Typical architecture of the evaporative cooling system.

4.4.2 Media of Evaporative Coolers

Evaporative coolers media used in gas turbine applications today is the wetted honeycomb-like medium to maximize evaporative surface area and cooling potential, as illustrated in Fig. 4.4. In this type of evaporative cooler, the evaporating medium is a saturated porous pad. Water is introduced through a header at the top of the media, sprays into the top of an inverted half-pipe, and is deflected downward onto a distribution pad on top of the media. Water drains through the distribution pad into the media, by gravity action downward through it, and wets enormous areas of media surface contacted by air passing through the cooler. An evaporative cooler can be either recirculating or non-recirculating. A recirculating evaporative cooler holds water in a reservoir and uses a pump to supply water to the header pipe. Water drains through the media into the reservoir below. Make-up water replacing evaporated water and water lost through blowdown flows into the reservoir. For a nonrecirculating cooler, new water is continuously introduced directly into the header from the water supply outside the cooler, passes once through the media, and is discarded. The water flow rate to the header should be the same for both types of coolers.

4.4.3 Operation Parameter

In determining the effect of the evaporative cooler on turbine performance, the dry-bulb temperature (T_{DB2}) of the air leaving the evaporative cooler must be known.

The outlet air temperature after cooling process as shown in Fig 4.5, can be calculated as:

$$T_{DB2} = T_{DB1} - \varepsilon(T_{DB1} - T_{WB1}) \qquad \textbf{(9)}$$

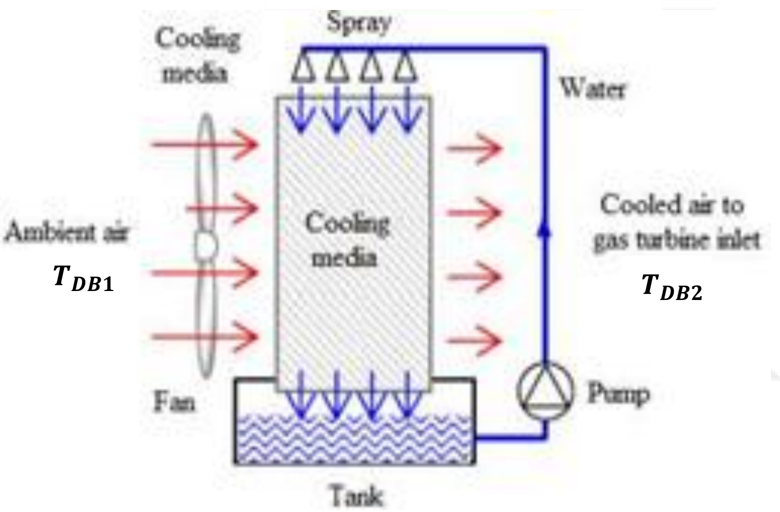

Figure 4.5: Schematic representation of the evaporative cooling media.

Where T_{DB2}, T_{DB1}, ε, T_{WB1} are dry-bulb temperature of the air outlet, dry-bulb temperature of the air inlet, evaporative cooling effectiveness and wet-bulb temperature of the inlet air respectively [4].

Evaporative cooling effectiveness, is the ratio of the actual air temperature drop to the maximum possible air side temperature drop.

In most applications, the relative humidity (RH) of the incoming air is known rather than wet-bulb temperature. The wet-bulb temperature can be found either from the psychometric chart or by using empirical expression as illustrated below in equation (10), [22].

$$T_{WB1} = T_{DB1}\, atan\left[0.151977(RH\% + 8.313659)^{1/2}\right] + \text{atan}(T_{DB1} + RH\%)$$

$$-\text{atan}(RH\% - 1.676331 + 0.00391838(RH)^{3/2}$$

$$\text{atan}(0.023101RH\%) - 4.68603 \quad \textbf{(10)}$$

Where T_{DB1} is in degree centigrade (C°) and RH is in percent (%).The cooling load associated with the evaporative cooling system result;

$$\dot{Q}_{CL} = \dot{m}_a C_{pa,avg.}(T_{DB1} - T_{DB2}) \quad \textbf{(11)}$$

Where $\dot{m}_a$ is the air mass flow rate and $C_{pa,avg.}$ is the specific heat of the dry air at constant pressure, determined as a function of the average temperature across the evaporative system, which is determined as:

$$C_{p,air}(T) = 1.04841 - \left(\frac{3.8371T}{10^4}\right) + \left(\frac{9.4537T^2}{10^7}\right) - \left(\frac{5.4903T^3}{10^{10}}\right) + \left(\frac{7.9298T^4}{10^{14}}\right) \quad \textbf{(12)}$$

Where T is the average temperature across the evaporative system [23].

4.4.4 Advantages and Disadvantages of Media of Evaporative Cooling System

The following list highlights some of the important advantages and disadvantages of evaporative inlet cooling system.

ADVANTAGES

1. Lowest capital cost.
2. Lowest operation and maintenance.
3. Water quality requirements less severe than other evaporative system.
4. Simple and reliable.
5. Quick delivery and installation time.
6. Operates as an air washer and clean the inlet air.

DISADVANTAGES

1. Uprates frequently require substantial duct modification.
2. Higher gas turbine inlet pressure drop than fogger system degrades output and efficiency when not in use.
3. Lower cooling effectiveness, by highly influenced to the outside wet bulb temperature.
4. Limitation on capacity improvement.

4.5 Absorption Cooling System

Most of industrial process uses a lot of thermal energy by burning fossil fuel to produce steam or heat for the purpose. After the processes, heat is rejected to the surrounding as waste. This waste heat can be converted to a useful refrigeration energy by using a heat operated refrigeration system, such as an absorption refrigeration cycle. Electricity purchased from utility companies for conventional vapor compression refrigerators can be reduced. The use of heat operated refrigeration systems help reduce problems related to global environmental, such as the so called greenhouse effect from CO2 emission from the combustion of fossil fuels in utility power plants.

Another difference between absorption systems and conventional vapor compression systems is the working fluid used. Most vapor compression systems commonly use chlorofluorocarbon refrigerants (CFCs), because of their thermo physical properties. It is through the restricted use of CFCs, due to depletion of the ozone layer that will make absorption systems more prominent. However, although absorption systems seem to provide many advantages, vapor compression systems still dominate all market sectors. In order to promote the use of absorption systems, further development is required to improve their performance and reduce cost.

The early development of an absorption cycle dates back to the 1700's. It was known that ice could be produced by an evaporation of pure water from a vessel contained within an evacuated container in the presence of sulfuric acid. In 1810, ice could be made from water in a vessel, which was connected to another vessel containing sulfuric acid. As the acid absorbed water vapor, causing a reduction of temperature, layers of ice were formed on the water surface. The major problems of this system were corrosion and leakage of air into the vacuum vessel. In 1859, Ferdinand Carre introduced a novel machine using water/ammonia as the working fluid. This machine took out a US patent in 1860. Machines based on this patent were used to make ice and store food. It was used as a basic design in the early age of refrigeration development. In the 1950's, a system using lithium bromide/water as the working fluid was introduced for industrial applications. A few years later, a double-effect absorption system was introduced and has been used as an industrial standard for a high performance heat-operated refrigeration cycle.[24]. Simple absorption chiller cycle is shown in Figure 4.6. The evaporator allows the refrigerant to evaporate and to be absorbed by the absorbent, a process that extracts heat from the source. The combined fluids then go to the generator, which is heated by the gas or steam, driving the refrigerant back out of the absorbent. The refrigerant then goes to the condenser to be cooled back down to a liquid, while the absorbent is pumped back to the absorber. The cooled refrigerant is released through an expansion valve into the evaporator, and the cycle repeats [*25*].

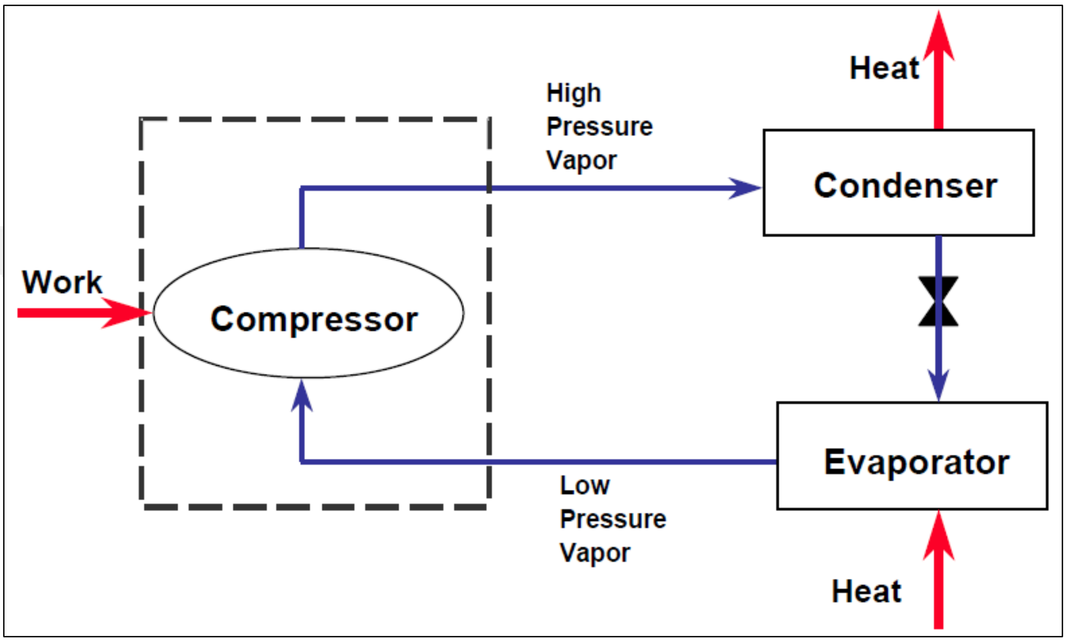

Figure 4.6: Simple absorption cycle diagram.

4.6 Principle of operation

The working fluid in an absorption refrigeration system is a binary solution consisting of refrigerant and absorbent. In Fig. 4.7(a), two evacuated vessels are connected to each other. The left vessel contains liquid refrigerant while the right vessel contains a binary solution of absorbent/refrigerant. The solution in the right vessel will absorb refrigerant vapor from the left vessel causing pressure to reduce. While the refrigerant vapor is being absorbed, the temperature of the remaining refrigerant will reduce as a result of its vaporization. This causes a refrigeration effect to occur inside the left vessel. At the same time, solution inside the right vessel becomes more dilute because of the higher content of refrigerant absorbed. This is called the "absorption process". Normally, the absorption process is an exothermic process, therefore, it must reject heat out to the surrounding in order to maintain its absorption capability. Whenever the solution cannot continue with the absorption process because of saturation of the refrigerant, the refrigerant must be separated out from the diluted solution. Heat is normally the key for this separation process. It is applied to the right vessel in order to dry the refrigerant from the solution as shown in Fig. 4.7 (b). The refrigerant vapor will be condensed by transferring heat to the surroundings. With these processes, the refrigeration effect can be produced by using heat energy. However, the cooling effect cannot be produced continuously as the process cannot be done simultaneously. Therefore, an absorption refrigeration cycle is a combination of these two processes as shown in Fig. 4.8.

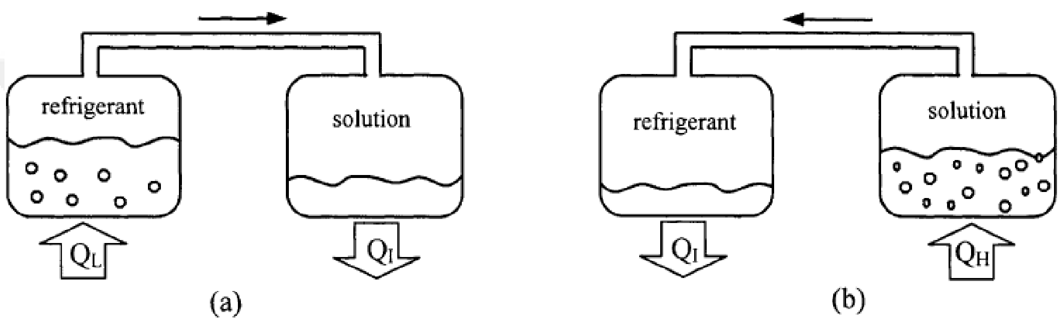

Figure 4.7: (a) Absorption process occurs in right vessel causing cooling effect in the other; (b) Refrigerant separation process occurs in the right vessel as a result of additional heat from outside heat source.

As the separation process occurs at a higher pressure than the absorption process, a circulation pump is required to circulate the solution. Coefficient of Performance of an absorption refrigeration system is obtained from;

$$\mathrm{COP} = \frac{Cooling\ capacity\ obtained\ at\ evaporator}{heat\ input\ for\ the\ generator + work\ input\ for\ the\ pump} \tag{13}$$

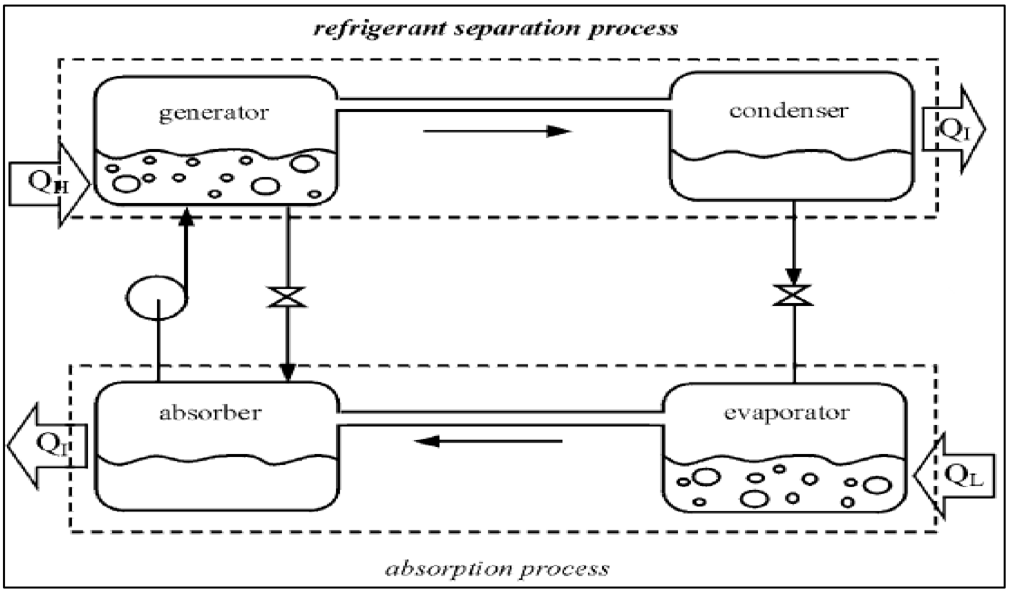

Figure 4.8: A continuous absorption refrigeration cycle composes of two processes mentioned in the earlier figure.

The work input for the pump is negligible relative to the heat input at the generator, therefore, the pump work is often neglected for the purposes of analysis [24].

4.7 Working Fluid for Absorption Refrigeration Systems

Performance of an absorption refrigeration systems is critically dependent on the chemical and thermodynamic properties of the working fluid [29]. A fundamental requirement of absorbent/refrigerant combination is that, in liquid phase, they must have a margin of miscibility within the operating temperature range of the cycle. The mixture should also be chemically stable, non-toxic, and non-explosive. In addition to these requirements, the following are desirable [24].

1. The elevation of boiling (the difference in boiling point between the pure refrigerant and the mixture at the same pressure) should be as large as possible.
2. Refrigerant should have high heat of vaporization and high concentration within the absorbent in order to maintain low circulation rate between the generator and the absorber per unit of cooling capacity.
3. Transport properties that influence heat and mass transfer, e.g., viscosity, thermal conductivity, and diffusion coefficient should be favorable.

4. Both refrigerant and absorbent should be non-corrosive, environmental friendly, and low-cost.

Many working fluids are suggested in literature. A survey of absorption fluids provided by Marcriss [27] suggests that, there are some 40 refrigerant compounds and 200 absorbent compounds available. However, the most common working fluids are Water / NH_3 and LiBr/water. Since the invention of an absorption refrigeration system water / NH_3 has been widely used for both cooling and heating purposes. Both NH_3 (refrigerant) and water (absorbent) are highly stable for a wide range of operating temperature and pressure. NH_3 has a high latent heat of vaporization, which is necessary for efficient performance of the system. It can be used for low temperature applications, as the freezing point of NH_3 is -77 C°. Since both NH_3 and water has volatility, the cycle requires a rectifier to strip away that normally evaporates with NH_3 [24].

4.8 Air Cooling Process

To get a clear understanding of the ambient air-cooling process, one often refers to a psychometric chart. Such a chart shown in Fig. 4.9 illustrates the path that air takes as it changes from ambient condition to the desired cooled state. By rejecting the sensible heat of air to the chilled water, the air temperature drops while its relative humidity continues to rise until its dew point is reached (point b in the figure). Any further cooling from this point will require removal of a much greater quantity of heat due to the latent heat of condensation of the water vapor that air contains in addition to its sensible heat. This process continues until it reaches the desired temperature at point c. The air-cooling process is plotted in Fig. 4.9 with heavy lines a b-c, and the sensible and latent heat loads are shown as d-c and a-d, respectively.

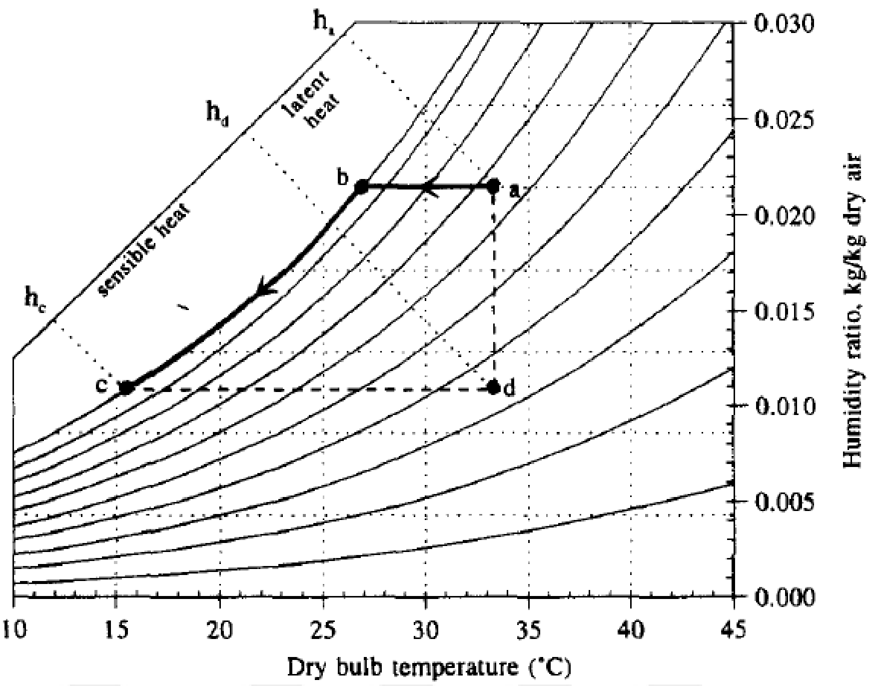

Figure 4.9: The air cooling process on psychometric chart.

Note that the final point of the cooling process corresponds to a relative humidity of 100% (i.e. Saturated) contrary to the 60%considered in an earlier study by Hufford [1 P. E. Hufford, Absorption chillers maximize cogeneration value, ASHRAE Trans.]. The RH corresponding to the ISO condition (i.e. 15°C and 60%RH) can be achieved in two ways. In the first option, only a part of the intake air passing through the heat exchanger coil is cooled to a much lower temperature and mixed with the rest of the air to reach the ISO condition; this will mean that very low temperature chilled water needs to be supplied by the absorption chiller to satisfy the low apparatus dew-point temperature requirement. The second option will require all the inlet air to be cooled to a much lower dew-point temperature and then have it relative humidity decreased from 100 to 60%by making use of a heater coil. Heating of the air for the purpose of attaining a RH of 60% (ISO condition) does not sound reasonable. In the first place, it will penalize the absorption chiller performance and add further complexity by integrating heating coils. Secondly, saturated air at 15°C will not affect the compression process since the air heats up during compression and there is hardly any possibility of moisture deposition on the compressor blades which can cause corrosion. The only factor that should be taken into account is an adequate separation of water droplets formed during the cooling and humidification processes, so that they are not carried along with the cooled air into the compressor. To prevent the moisture from entering the compressor, a filtration/separation system can be installed immediately before the compressor inlet [28].

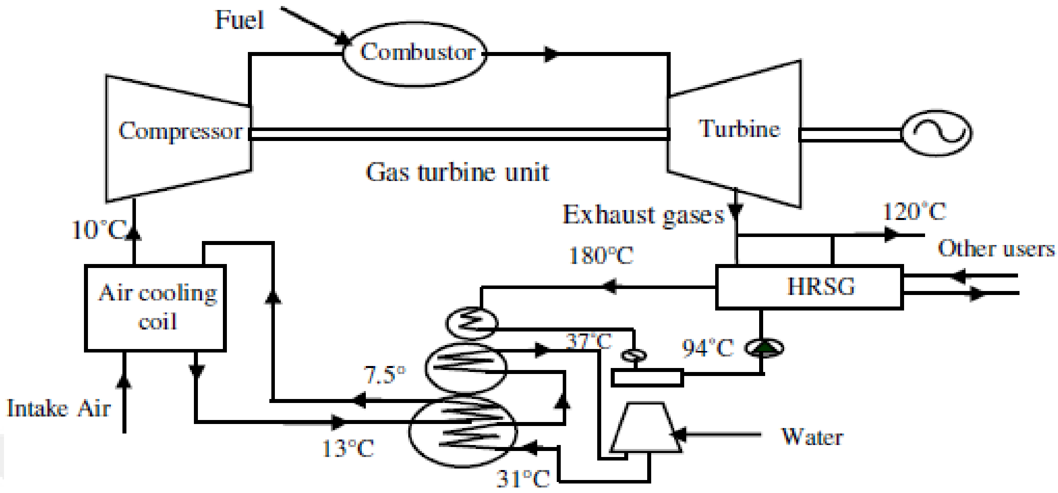

Figure 4.10: A schematic diagram for Absorption system with gas turbine.

The total cooling load (Qc) is composed of the heat required to reduce the temperature from its initial ambient condition to the desired cooled state, i.e. the sensible heat of the air (q_s) and the heat required to condense the moisture in the air or the latent heat (q_l). So Q_C can be computed as:

$$Q_C = m_{air}(q_s - q_l) \quad \textbf{(14)}$$

$$q_s = h_d - h_c \quad \textbf{(15)}$$

$$q_l = h_{fg}(\omega_a - \omega_b) \quad \textbf{(16)}$$

h_d and h_c are the enthalpies of air at points d and c, as shown in Fig. 4.9 and $h_{fg}$is the latent heat of water at T_{WB}.

CHAPTER FİVE

CALCULATIONS AND RESULTS

5.1 Gas Turbine Cycle

Figure 5.1 shows the considered ignition single shaft gas turbine cycle. The model utilized to reproduce the air thermodynamic states from the gulf stage to the last, debilitate gasses stage. Weight and temperature counts for every point are likewise decided, as appeared in Figure 5.1.

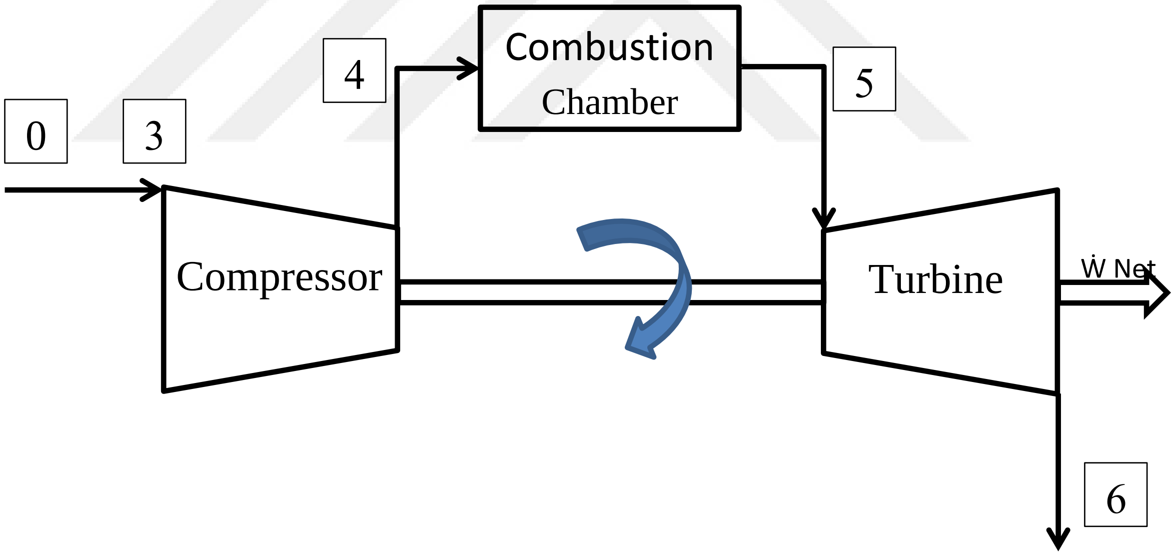

Figure 5.1: Schematic of the typical gas turbine cycle.

5.2 Governing Equations

Steady state continuity equation, first and second low of thermodynamic equations are used to pressure, temperature, work done, heat add and power output for each point of the system.

The compressor inlet pressure is equal to ambient pressure once that pressure drop at inlet and exhaust ducts are neglected, resulting that:

$$P_0 = P_3 \qquad (\mathbf{17})$$

The pressure of the air leaving the compressor (P4) is calculated as:

$$P_4 = p_r . P_3 \qquad (\mathbf{18})$$

Where is the p_r compressor pressure ratio.

Using the polytropic relations for ideal gas and knowing the isentropic efficiency of the compressor the discharge temperature (T_4) can be calculated as:

$$T_4 = \frac{T_3}{\eta_c}\left[\left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] + T_3 \qquad (\mathbf{19})$$

Where η_c is the compressor isentropic efficiency and γ is the specific heat ratio.

The compressor work can be estimated using the first law of thermodynamic as follows:

$$\dot{W}_C = \dot{m}_a . C_{pa,avg}(T_4 - T_3) \qquad (\mathbf{20})$$

Where $\dot{m}_a$ is the air mass flow rate and $C_{pa,avg}$ is the specific heat of the dry air at constant pressure, determined as function of the average temperature across the compressor, Eq. (12).

Expecting a pre-characterized combustor weight drop ($\Delta P_{Combustr}$), the burning chamber release weight (P_5) can be ascertained as:

$$P_5 = P_4 - \Delta P_{Combustor} \qquad (\mathbf{21})$$

The heat conveyed by burning chamber is resolved from vitality parity as:

$$\dot{Q}_{in} = \dot{m}_T . C_{pg,avg} . (T_5 - T_4) \qquad \textbf{(22)}$$

Where $C_{pg,avg}$ is the flue gas specific heat calculated as function of the average temperature across the combustion chamber, as:

$$C_{pg,avg} = 0.991615 + \left(\frac{6.99703T}{10^5}\right) + \left(\frac{2.7129T^2}{10^7}\right) - \left(\frac{1.22442T^3}{10^{10}}\right) \qquad \textbf{(23)}$$

By knowing the fuel gas heat value (FHV), the natural gas mass flow rate is defined as:

$$\dot{m}_f = \frac{Q_{in}/FHV}{\eta_{Combustor}} \qquad \mathbf{(24)}$$

Where $\eta_{Combustor}$ is the combustion chamber efficiency.

The turbine release temperature can be composed as:

$$T_6 = T_5 - \eta_t . T_5 \left[1 - \left(\frac{1}{P_5/P_6}\right)^{\frac{\gamma-1}{\gamma}}\right] \qquad \mathbf{(25)}$$

Where η_t is the turbine isentropic efficiency and P_6 is the ambient pressure.

Hence the turbine work done rate is equal to:

$$\dot{W}_t = \dot{m}_T . C_{pg,avg}(T_5 - T_6) \qquad \mathbf{(26)}$$

Where $\dot{m}_T$ is the total mass flow rate, it is composed for fuel and air mass flow rate:

$$\dot{m}_T = \dot{m}_a + \dot{m}_f \qquad \mathbf{(27)}$$

Finally, the net work done rate gained from the gas turbine is done by:

$$\dot{W}_{Net} = \dot{W}_T - \dot{W}_C \qquad \mathbf{(28)}$$

The thermal efficiency of the gas turbine is determined by the following equation:

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \qquad \mathbf{(29)}$$

5.3 Assumptions

To solve the above equations correctly, there are some assumptions deal with it:

1. Air and hot gases considered as ideal gas.
2. The variation of pressure at compressor inlet air duct and turbine exhaust is 1.2%.

3. The variation in air Relative humidity at different temperature is negligible.
4. Isentropic efficiency of compressor is assumed (85%), (This value is significantly relies on upon the configuration of the compressor. Very much composed compressor have isentropic efficiencies that extent from 80 to 90 %) [15].
5. Isentropic efficiency of the turbine is assumed (87%), (This quality is enormously relies on upon the configuration of the compressor. Very much outlined compressor have isentropic efficiencies that extent from 80 to 90 %) [15].
6. Changes in potential energy and kinetic energy are not considered.
7. Friction losses is negligible and the system is adiabatic.
8. Efficiency value between generator and turbine is 0.97.

5.4 Operation Data

Real data are collected from Duhok Power Station after contacted with the manager of the station and head of operation department. This data was for period of one year with data sheet as in table 3.2, by taking continuous reading six times every day, once every four hours. Data sheet contains parameters which indicate the real operation data for the power generation unit GE gas turbine MS9001E, this parameters are listed below which are used in calculations.

1. Generator power output [MW].
2. Intake Air temperature T3 [C°].
3. Compressor discharge temperature T4 [C°].
4. Intake Air pressure P3 [kPa].
5. Compressor discharge pressure P4 [kPa].
6. Fuel consumption $\dot{m}_f$ [Kg/s].
7. Mass of inlet air $\dot{m}_{fa}$ [Kg/s].
8. Exhaust temperature T6 [C°].
9. Energy power output [MWh].
10. Fuel used is diesel and Low Heat Value is 43,448 [kJ/kg].

5.5 Calculations

Initially, and before adding any cooling system, a Base-Case was compered with ISO conditions (T_{inlet} = 15 C° and ϕ = 60 %) as shown in Fig. 5.1. To make sure that we are walk in the right way, and before adding any cooling system, an comparison between real and theoretical "Isentropic" conditions. Performance should be verified to see the deviation between them, then observing the effect of adding cooling system.

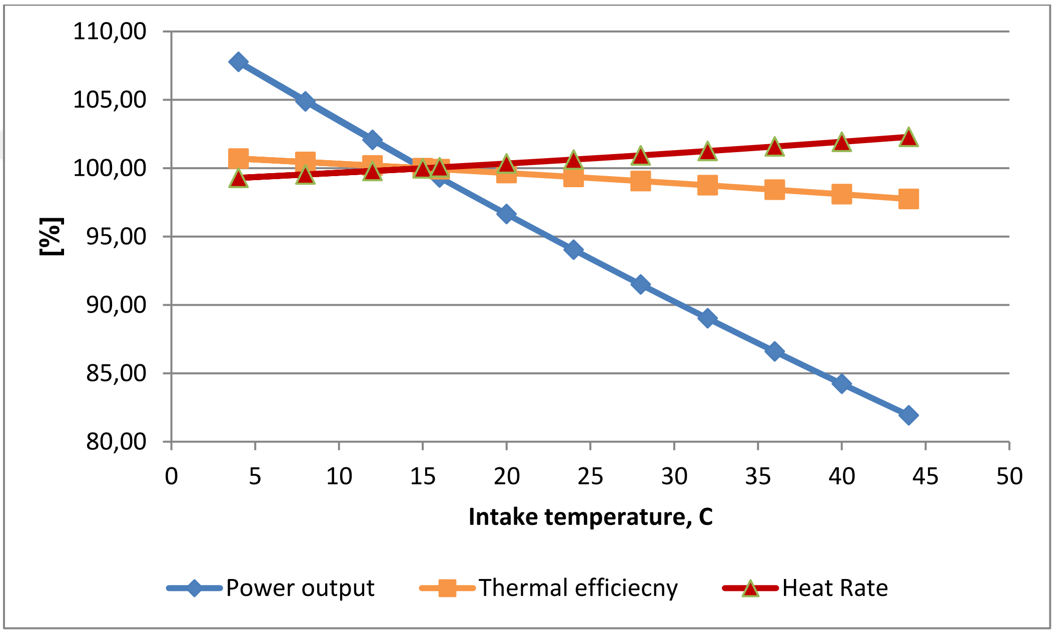

Figure 5.2: Impact of inlet temperature on the gas turbine performance with ISO conditions.

5.5.1 Compressor Power Input Calculation

The isentropic efficiency of a compressor is characterized as the proportion of the work information required to raise the weight of a gas to a predetermined worth in isentropic way to the genuine work input:

$$\xi_C = \frac{\text{Isentropic compressor work}}{\text{Actual compressor work}} = \frac{w_s}{w_a} \qquad \textbf{(30)}$$

Observe that the isentropic compressor effectiveness is characterized with the isentropic work contribution to the numerator rather than in the denominator. This is on the grounds that ws is a littler amount than wa, and this definition keeps proficiency from getting to be more noteworthy than 100 percent, which would dishonestly suggest that the real compressors performed superior to the isentropic ones. Likewise see that the delta conditions and the way out weight of the gas are the same for both the genuine and the isentropic compressor.

At the point when the progressions in dynamic and potential energies of the gas being compacted are insignificant, the work contribution to an adiabatic compressor gets to be equivalent to the change in enthalpy, and efficiency's equation for this case becomes:

$$\xi_C \cong \frac{h_{4s}-h_3}{h_{4a}-h_3} \tag{31}$$

where $h4a$ and $h4s$ are the enthalpy values at the way out state for genuine and isentropic pressure forms, separately, as outlined in Figure 5.2.

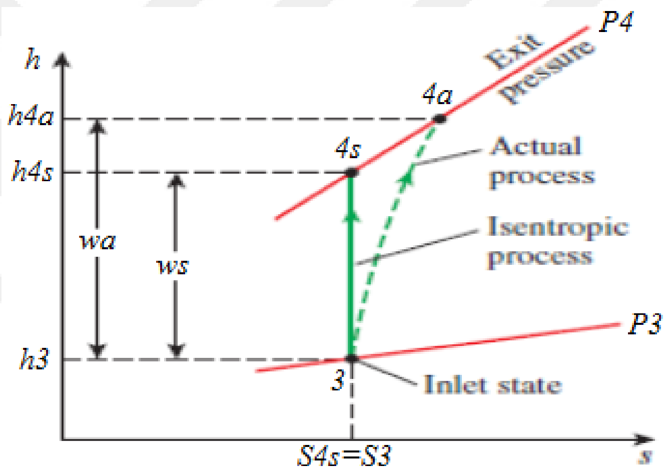

Figure 5.3: The h-s graph of the real and isentropic processes of an adiabatic compressor.

Most important performance is work required for compressed air from specific pressure and temperature to a new specific pressure and temperature at a steady rate of mass. The genuine required force contribution to the compressor is resolved from the vitality adjusted for unfaltering stream gadgets:

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_a h_3 + \dot{W}_{a,in} = \dot{m}_a h_{4a}$$

$$\dot{W}_{a,in} = \dot{m}_a (h_{4a} - h_3)$$

$$\dot{W}_{a,in} \cong \dot{m}_a * Cp_{a,avg.} * (T_{4a} - T_3)$$

Notice that we use actual enthalpy or actual exit temperature and particular warmth of the air at consistent pressure, decided as an element of the normal temperature over the compressor to calculate real power input. To determining the isentropic power input, h_{4s} *or* T_{4s} must be used.

By using isentropic efficiency of compressor Eq. (5.31) we could calculate h_{4s} *or* T_{4s}, then calculate isentropic power input for compressor.

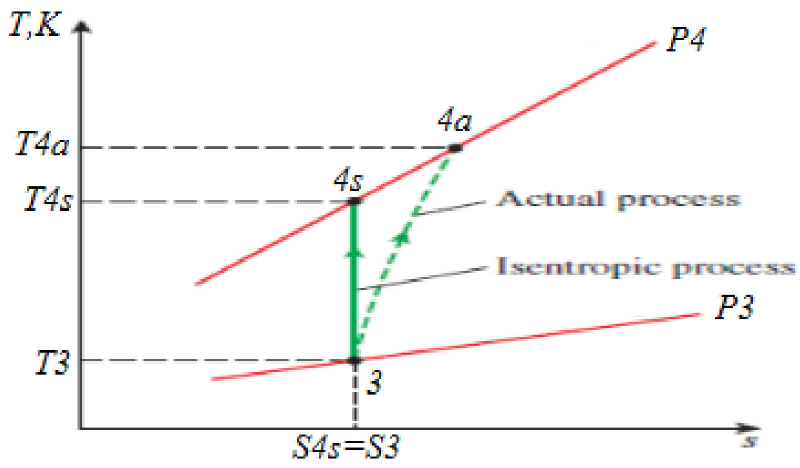

Figure 5.4: The T-s graph of the real and isentropic processes of an adiabatic compressor.

5.5.2 Turbine Power Output Calculation

For a turbine work enduring operation, the channel condition of the working liquid and the fumes weight are settled. Accordingly, the perfect procedure for an adiabatic turbine is an isentropic procedure between the channel state and the fumes weight. The sought yield of a turbine is the work delivered, and the isentropic effectiveness of a turbine is characterized as the proportion of the genuine work yield of the turbine to the work yield that would be accomplished if the procedure between the channel state and the way out weight were isentropic:

$$\xi_T = \frac{Actual\ turbine\ work}{Isentropic\ turbine\ work} = \frac{w_a}{w_s} \qquad \textbf{(32)}$$

Normally the progressions in dynamic and potential energies connected with a liquid stream moving through a turbine are little in respect to the adjustment in enthalpy and can be dismissed. At that point, the work yield of an adiabatic turbine essentially turns into the adjustment in enthalpy:

$$\xi_T \cong \frac{h_5 - h_{6a}}{h_5 - h_{6s}} \quad \textbf{(33)}$$

where $h6a$ and $h6s$ are the enthalpy values at the way out state for genuine and isentropic forms, separately. The proficiency esteem enormously relies on upon the configuration of the singular segments that make up the turbine. Very much outlined, extensive turbines have isentropic efficiencies above 90 percent. For little turbines, notwithstanding, it might drop indeed, even beneath 70 percent. The estimation of the isentropic proficiency of a turbine is controlled by measuring the genuine work yield of the turbine and by computing the isentropic work yield for the deliberate bay conditions and the way out weight .This quality can then be utilized helpfully as a part of the configuration of force plants, as in Fig. 5.4.

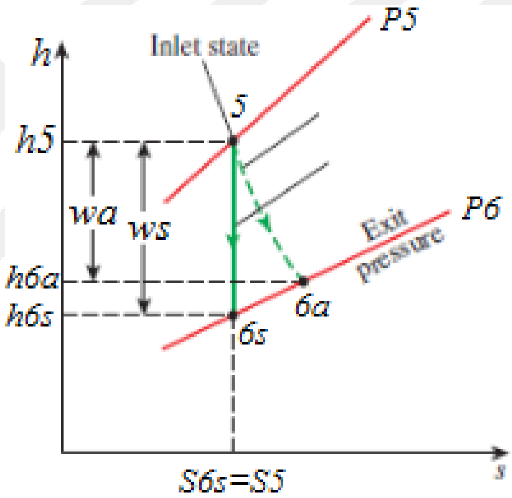

Figure 5.5: h-s diagram for the actual and isentropic processes of an adiabatic turbine.

To calculation real turbine power output, from the energy balance for steady flow,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_T h_5 + \dot{W}_{Ta,out} = \dot{m}_a \dot{h}_{6a}$$

$$\dot{W}_{Ta,out} = \dot{m}_T (h_5 - h_{6a})$$

$$\dot{W}_{T,out} \cong \dot{m}_T * Cp_{g,avg.} * (T_5 - T_{6a})$$

The most important parameter which is the temperature of the hot gases leaves from the combustion chamber and enters the turbine (T_5) is unavailable from the data sheet, so we use equations (5.34&5.35) listed below to calculate it.

$$\dot{m}_f = \frac{\dot{Q}_{in}/_{LHV}}{\xi_{Combustor}} \tag{34}$$

Where $\xi_{Combustor}$ is the combustors chamber efficiency which is equal to (0.99).

$$\dot{Q}_{in} = \dot{m}_a . C_{pg,avg.} . (T_5 - T_4) \tag{35}$$

Then we use (T_5) temperature to find real power output and thermal efficiency from the turbine. By using isentropic efficiency of the turbine we calculate (T6s & h6s) then calculate isentropic power output and thermal efficiency.

$$\dot{W}_{Tisn.,out} = \dot{m}_T (h_5 - h_{6s})$$

$$\dot{W}_{Tisn.,out} \cong \dot{m}_T * Cp_{g,avg.} * (T_5 - T_{6s})$$

5.6 Monthly Temperature Distribution in Duhok City

By using DGPS's data sheet, one from the operating parameter is ambient temperature, which is very important indication to see the variety of the air intake temperature throughout the year. Fig 5.5 clarify the monthly temperature average values during the year 2013, which is created from the data sheet of the DGPS.

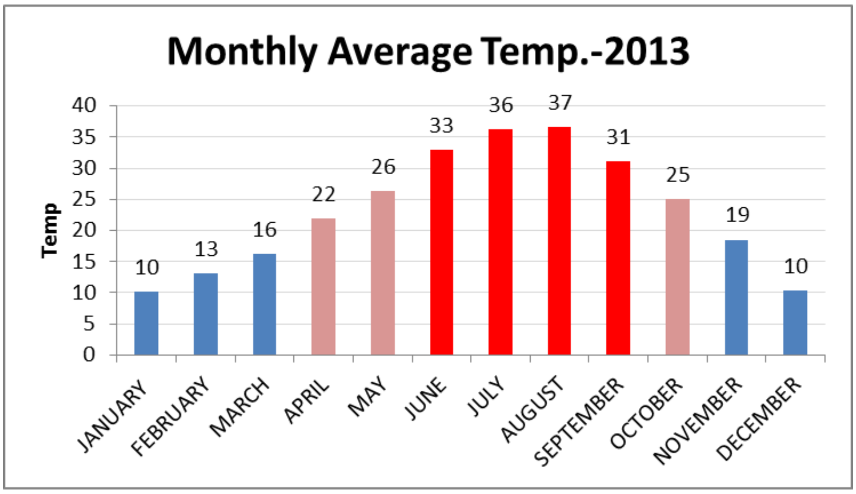

Figure 5.6: Monthly average ambient temperature distribution -2013.

As we can see in the Fig. 5.6, that the ambient temperature is in low zone during first and fourth quarter of the year, while it is be at intermediate and high zone during second and third quarter of the year, this increasing in the intake air temperature significantly effect on the unit's performance output. For this reason we choose a two different cooling methods, that will use them in theoretically manner to improver power generating unit and try to increase the power output in that time electricity demand increased rapidly and the great part of the energy added wasted to atmosphere "The period of using cooling system will be from May to October".

5.7 Gas Turbine Inlet Air Cooling System

Gas turbine air bay cooling is valuable strategy for expanding power yield and warm productivity for districts where critical force request and most elevated power costs happen particularly amid the warm months.

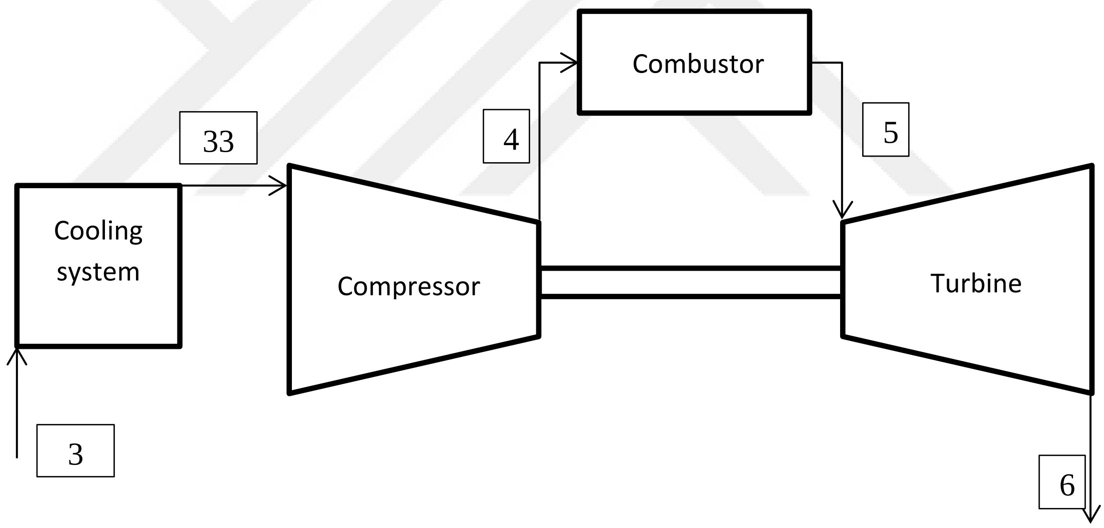

Figure 5.7: Schematic of the gas turbine cycle with cooling system.

In this study, two basic cooling system will integrated, first is evaporative cooling, which make utilization of vanishing of water to decrease the gas turbine's channel temperature. The second is assimilation cooling framework, which utilizes to cool the delta air through cooling medium moves through a warmth exchanger situated in the bay channel to expel heat from the gulf air.

5.7.1 Evaporative Cooling System

The evaporative cooling structure is most appropriated to hot and dry regions, since it utilizes the dormant warmth of vaporization to cool incorporating temperature from the dry-handle to wet-globule temperature. Essential media sorts of evaporative coolers use a wetted honeycomb-like medium to enlarge evaporative surface district and cooling potential, as spoke to in Fig.6.5. Regularly this cooling apparatus is set after the air channel system Fig. 5.6. The cove air temperature in the wake of cooling method (T_33), see Fig. 5.6, can be registered using eq. (9). Where T_b3 is the dry-globule temperature, T_w3 is the wet-knob temperature and ε is evaporative cooling viability. Wet-knob temperature is computed by suing eq. (10).

Where T is inlet air dry –bulb temperature and HR is relative humidity, which we use it as (40%). The cooling load associated with the evaporative cooling system calculated form eq. (11), and the specific heat of air at constant pressure $C_{pa,avg.}$, determined as a function of the average temperature across the evaporative system eq. (12).

The feasible temperature is confined by the wet-bulb temperature of this method. The evaporative water mass flow associated with the evaporative cooling can be calculated by eq. (8). After adding evaporative cooling system a set of equations and assumptions are sued to re-calculate parameters and performance for the new conditions

5.7.2 Absorption Cooling System

One more method to give gas turbine admission air cooling is the assimilation chiller component, as illustrated in Fig. 5.7. The heat required to chilled water is normally acquired by the recuperation of the warmth from turbine deplete gasses, and the chilled water is gone through a warmth exchanger to cool the encompassing air temperature.

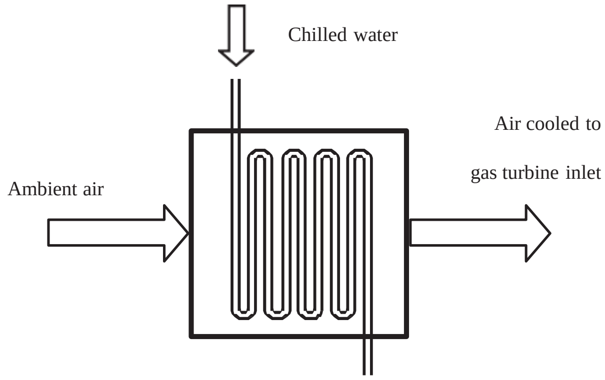

Figure 5.8: Schematic representation of the chiller coil.

Absorption system in power plants can utilize lithium-bromide or ammonia-water mix. At the point when the ingestion chiller is utilized, the compressor bay air temperature is free of wet-knob temperature, however there is a base adequate worth forced by the compressor icing development hazard. At this work, a temperature of 10 C° was received. In this cycle, the cooling load expelled from the air streaming at encompassing conditions into the force plant can be ascertained applying first low of thermodynamic as follows:

$$\dot{Q}_{CL} = \dot{m}_a * \left[(h_3 - h_{33}) - h_{\omega,33} * (\omega_3 - \omega_{33})\right] \quad \textbf{(36)}$$

Where h_3 and h_{33} are the enthalpy in the inlet and outlet of the system respectively. Water bundle present noticeable all around will be dense in the warmth exchanger and might be utilized at the plant as a part of alternate procedure. This amount of water can be evaluated by:

$$\dot{m}_w = \dot{m}_a * (\omega_3 - \omega_{33}) \quad \textbf{(37)}$$

Same equations which are be used in Evaporative cooling system to calculated parameters and performance will be used in Absorption system. One change between first and second cooling systems is the inlet temperature of the air, where in the first one inlet temperature is changeable while in the second one inlet is fixed at design temperature.

5.8 Results

After we verifying the performance of the power generator unit at ISO conditions as illustrated in Fig. 6.1, a comparison between real and Isentropic conditions verified for compressor, turbine and gen-set thermal efficiency to sight the deviation between them as see in Fig. 5.8, Fig. 5.9 and Fig. 5.10.

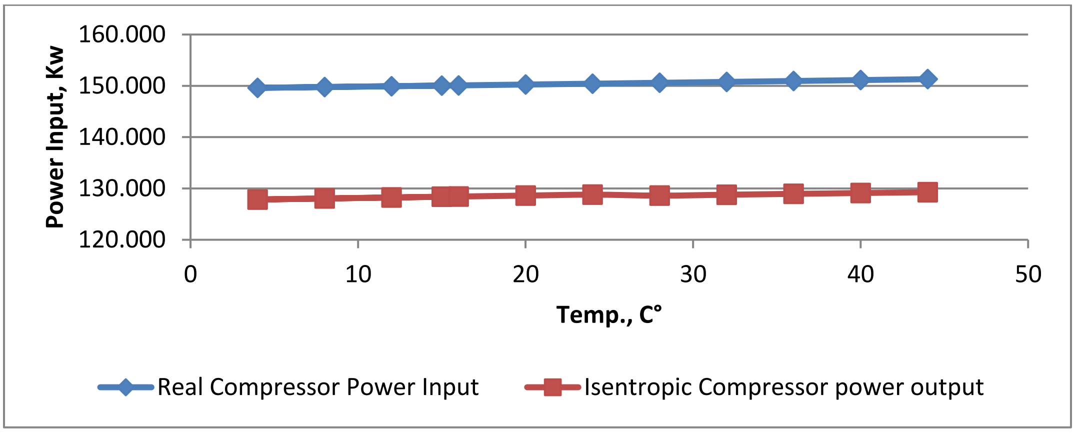

Figure 5.9: Real and Isentropic Compressor power Input at ISO conditions.

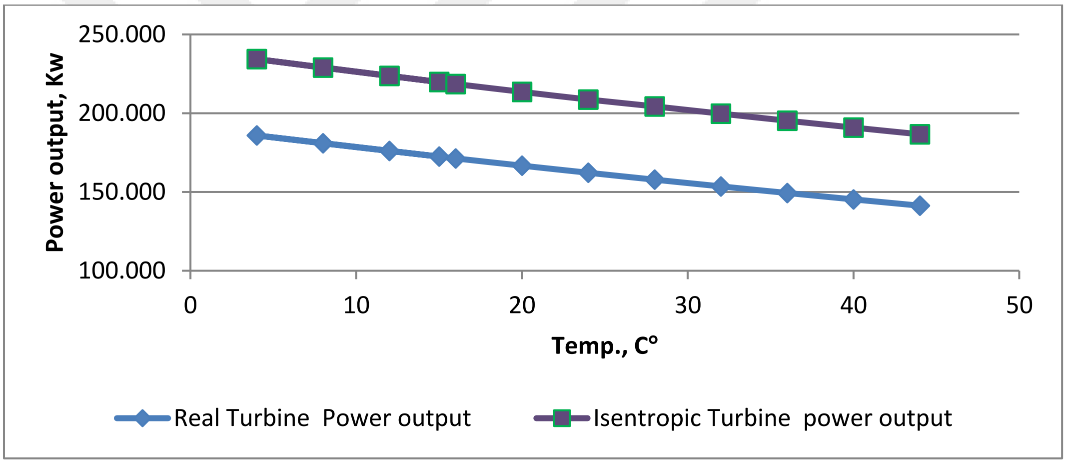

Figure 5.10: Real and Isentropic Turbine Power Output at ISO conditions.

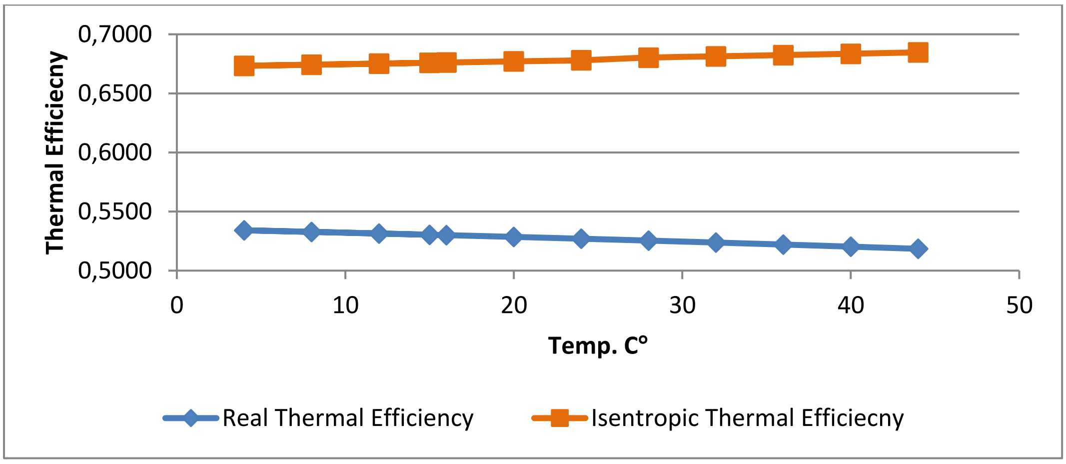

Figure 5.11: Real and Isentropic Gen-set Thermal Efficiency at ISO conditions.

5.8.1 ISO Conditions Results

All the gas turbine designer and manufacturers are consider ISO conditions, but as we know gas turbine unites are spread all over the world with different environmental conditions, some places are arid "hot and dry", some of them are tropical "hot and wet" and others places are between and among. This variable in environmental conditions effect in a great way on the unit performance.

So it's necessary to see this variations under different conditions and also the effect of adding cooling system to the unit.

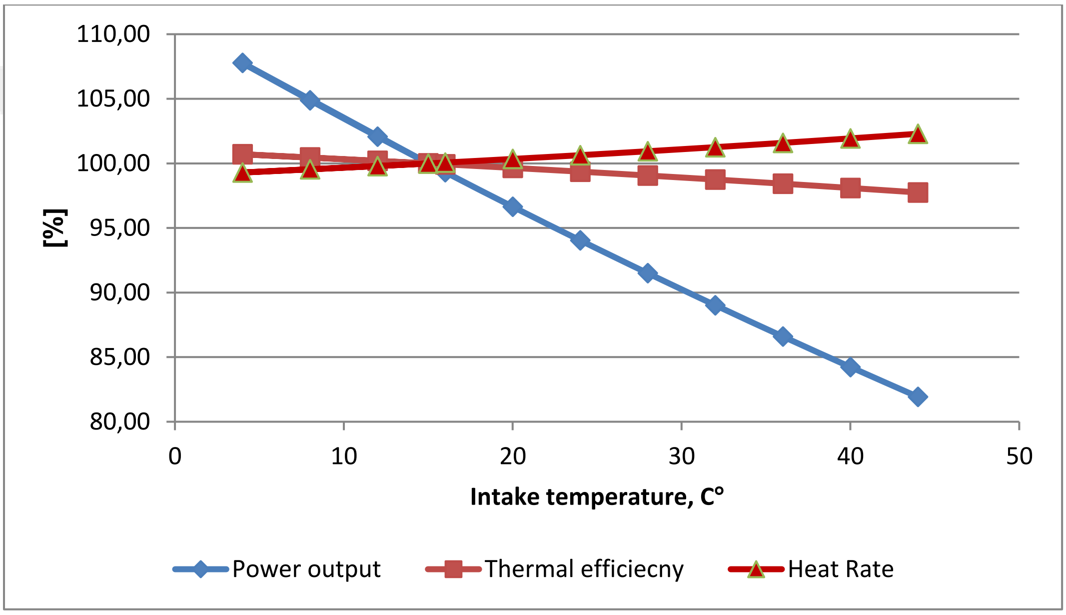

Figure 5.12: Effect of air intake temperature comparison with ISO conditions.

As we can see in Fig. 5.11, turbine loss near to 20% from its power output when ambient temperature rich to near 50 C°. To avoid the loss, it is necessary to add cooling system in the air inlet, so we can correct the air inlet temperature and try to decrease it as much as possible. This could be done by using either evaporative cooling system or absorption chilling system.

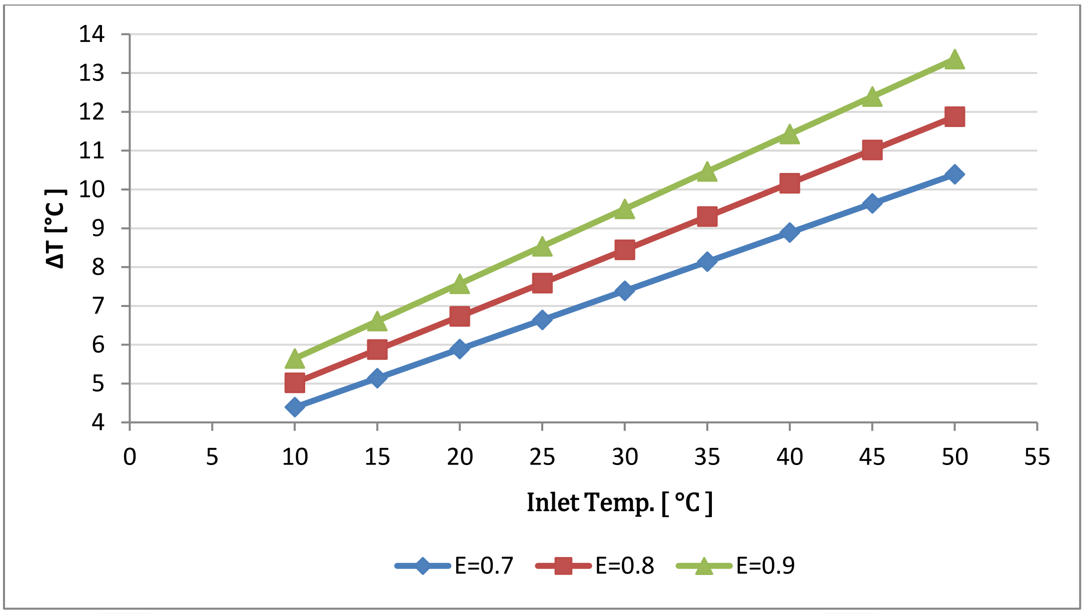

Figure 5.13: Effect of evaporative cooling effectiveness on the temperature drop.

First of all we will apply evaporative cooling system with different cooling effectiveness, and see the effect of that on temperature drop, power output and thermal efficiency at (ISO) conditions.

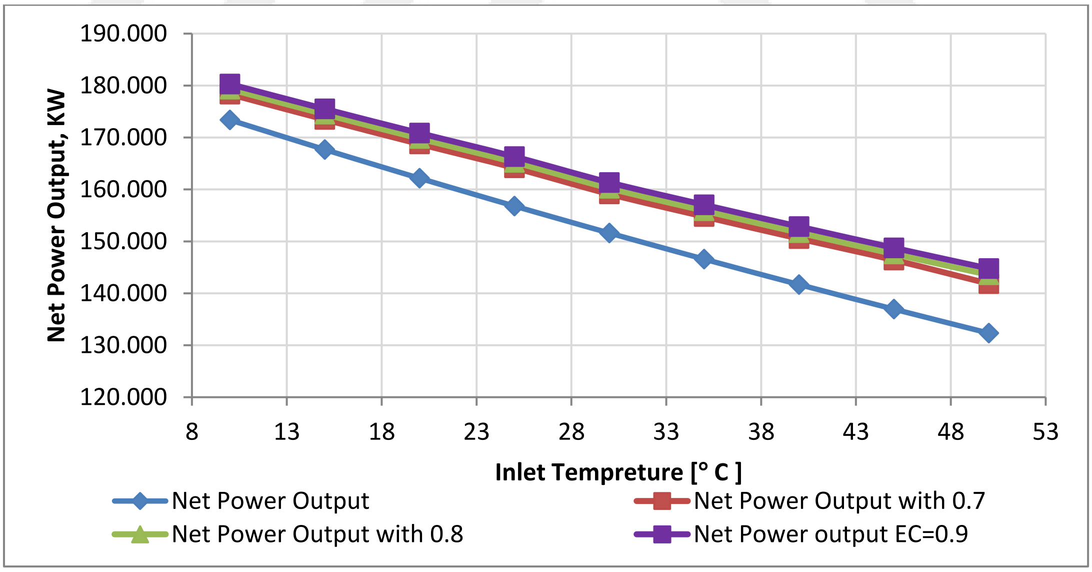

Figure 5.14: Effect of evaporative cooling effectiveness on power output.

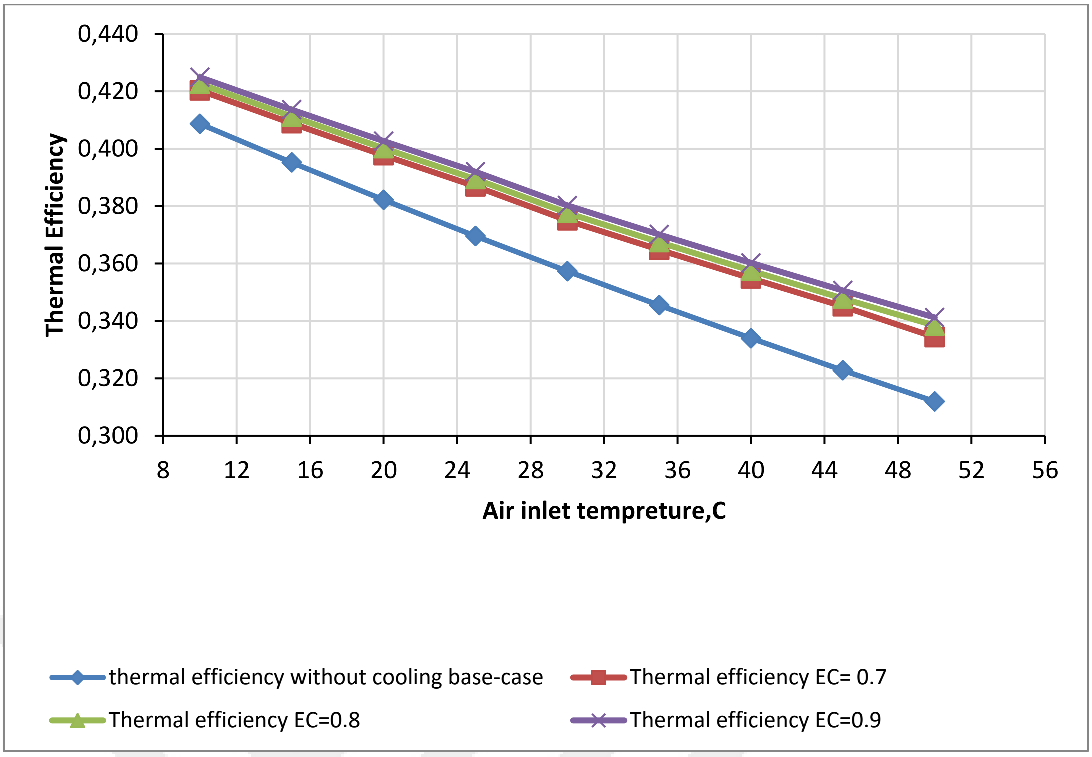

Figure 5.15: Effect of evaporative cooling effectiveness on the thermal efficiency.

Secondly we will apply absorption cooling system with different output temperature (10, 12, 14) C°, and see the effect of that on power output and thermal efficiency.

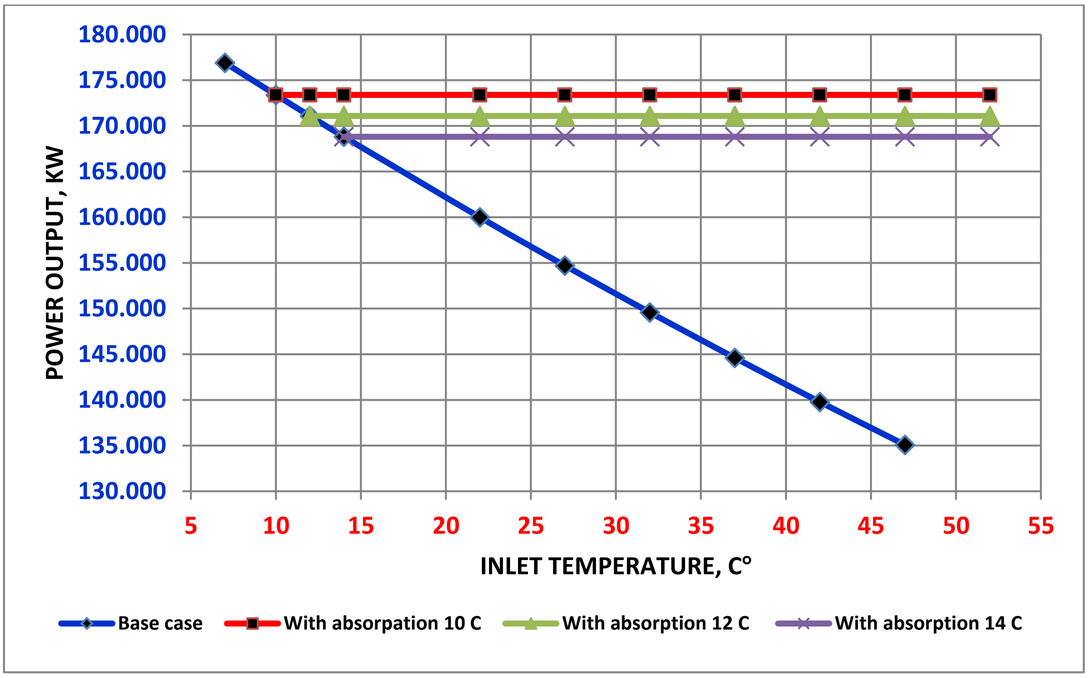

Figure 5.16: Effect of ambient intake temperature on the gas turbine power output with and without absorption chiller cooling system.

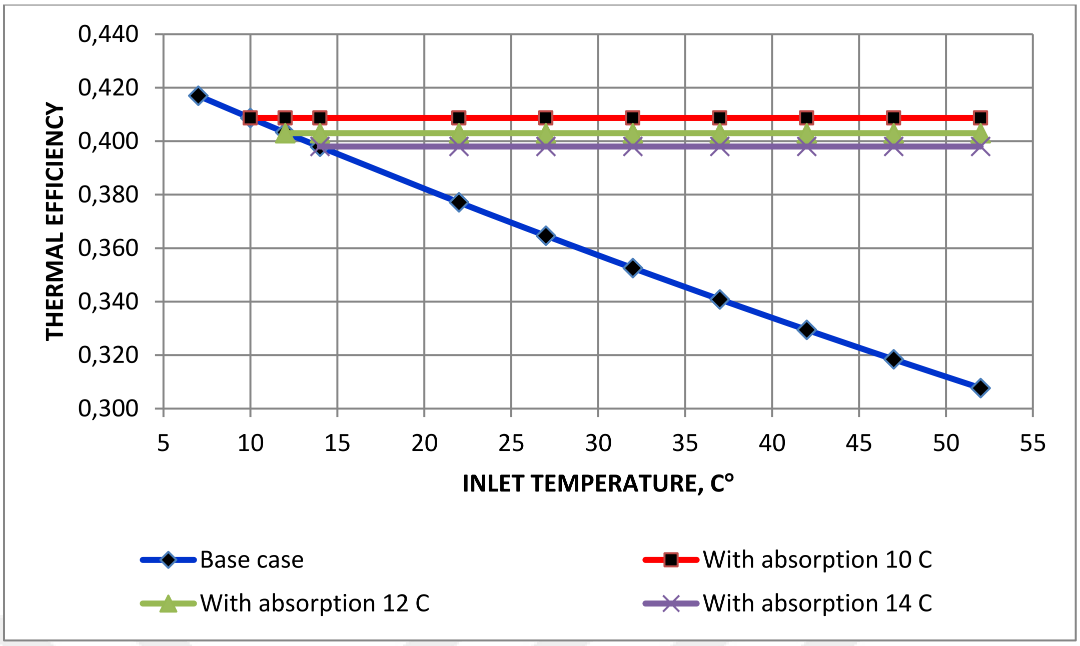

Figure 5.17: Effect of ambient temperature on the gas turbine thermal efficiency with and without absorption chiller cooling system.

5.8.2 Different Environmental Conditions Results

Because of the third and fourth quarter "May to October" are in the intermediate and high zone of the air intake temperature, so we will choose this period and make our calculation in two cases, first before adding cooling system and second after adding two types cooling system. We will discussed after that the effect of adding cooling system from the operating and economical view. In each case we will calculated the compressor power input, turbine power output and thermal efficiency. We make our calculation every month separately from May to October, at every month we get an monthly compressor power input, turbine power output, thermal efficiency, energy output and fuel consumption. This calculations covered three cases;

1. Base case: Normal operation without any adding cooling system.
2. Evaporative cooling: Operation with add evaporative cooling system with temperature setting is (14 C°).
3. Absorption cooling: Operation with add absorption cooling system with temperature setting is (10 C°).

Table 5.1: Different Operation Mode and effect of adding cooling systems.

Months	Operation Mode		
	BC- Compressor power input [Kw]	EC- Compressor power input [Kw]	AS- Compressor power input [Kw]
May	123,738	116,310	112,741
June	124,580	115,453	109,698
July	131,711	123,680	116,378
August	131,383	123,966	90,455
September	121,383	116,747	111,563
October	113,997	105,960	103,717
Months	BC- Turbine power output [Kw]	EC- Turbine power output [Kw]	AS- Turbine power output [Kw]
May	57,913	81,149	81,090
June	57,105	75,934	78,598
July	74,621	100,158	106,525
August	79,635	109,200	112,510
September	56,610	79,864	82,047
October	45,328	67,090	68,345
Month	BC- Thermal Efficiency	EC- Thermal Efficiency	AS- Thermal Efficiency
May	0.261	0.348	0.348
June	0.259	0.337	0.337
July	0.278	0.363	0.369
August	0.287	0.373	0.365
September	0.248	0.344	0.338
October	0.228	0.352	0.335

As we can see in table 5.1, there are observed decreasing in the compressor power input, which is lead to increasing in turbine power output and thermal efficiency, this saving and increasing in energy output is offset by a further increase in amount of fuel used, which is necessary to keep the turbine inlet temperature gases constant. Table 5.2 illustrate the effect of this increasing and the amount of energy output.

Table 5.2: Increasing in energy output and fuel consumption.

Months	Operation Mode	Energy Output [KWH / Month]	Fuel Consumption [Litter / Month]
May	Base Case	40,920,000	13,734,424
	Operation with Evaporative	48,803,000	13,898,089
	Operation with Absorption	50,276,000	14,350,987
June	Base Case	38,000,000	13,111,398
	Operation with Evaporative	45,560,000	13,409,145
	Operation with Absorption	47,159,000	14,120,179
July	Base Case	53,415,000	16,449,032
	Operation with Evaporative	62,098,000	16,893,400
	Operation with Absorption	64,299,000	17,805,199
August	Base Case	58,450,000	17,699,000
	Operation with Evaporative	68,837,000	18,151,297
	Operation with Absorption	69,756,000	19,082,517
September	Base Case	33,966,000	13,481,047
	Operation with Evaporative	47,919,000	13,747,350
	Operation with Absorption	49,288,000	14,381,529
October	Base Case	14,505,000	6,348,000
	Operation with Evaporative	21,504,000	6,480,297
	Operation with Absorption	21,870,000	6,575,974

5.9 Economic Performance of the Inlet Air-Cooling Systems

So as to assess the plausibility of a cooling framework coupled to a gas turbine plant, the execution of the plant is analyzed with and without the cooling framework. As a rule, the net force yield of a complete framework is equal to compressor power input and cooling system electricity consumed subtracted from turbine power output.

This analysis considers the power gain ratio (PGR), a broad term suggested that takes into account the operation parameters of the GT and the associated cooling system.

$$PGR = \frac{\dot{W}_{net,with\ cooling} - \dot{W}_{net,without\ cooling}}{\dot{W}_{net,without\ cooling}} * 100\% \quad \textbf{(38)}$$

For a stand-alone GT, PGR = 0. Thus, PGR gives the percentage enhancement in power generation by the coupled system.

The thermal change factor (TEC) proposed is defined as

$$TEC = \frac{\eta_{with\ cooling} - \eta_{without\ cooling}}{\eta_{without\ cooling}} * 100\% \qquad \textbf{(39)}$$

Both PGR and TEC can be effortlessly utilized to survey the adjustments in the framework execution, yet are not adequate for a complete assessment of the cooling strategy. To investigate the economic feasibility of retrofitting a gas turbine plant with an intake cooling system, the total cost of cooling system should be determined [29]. From the plant owner and owner's and administrator's perspective, an imperative element for choosing which framework ought to be chosen for admission air-cooling, aside from the addition in power era that has been displayed in the past segment, is the monetary execution of the speculation. Keeping in mind the end goal to evaluate the financial execution of the gas turbine air-cooling framework under thought, the essential monetary pointers of the speculations, to be specific, the payback time frame (PB) and the aggregate expense of the incremental power generation (ICOE) [3]. The particular expenses for every cooling framework utilized computations from the local market survey are submitted in Table 5.3.

Table 5.3: Estimated costs of every cooling system.

Cooling Method	Estimated costs $/Kw
Evaporative system	67.5
Absorption system	281.5

5.9.1 Electricity Selling Prices

As it's known for all, the electricity prices are varying from day and night, summer and winter, also types of use in other words domestic and industrial, the incremental cost of cost of KWh and investment payback period, were calculated on the basis of sale of electricity at present Iraqi Kurdistan regain prices. Mean electricity selling price is (0.031 $/KWH).

5.9.2 Electricity Generation Cost

1. Cost of used fuel

Mean fuel cost is (0.35 $/L).

2. Operation and Maintenance costs

Mean Operation and Maintenance costs is (0.0055 $/ KWH) [13].

First of all we should calculate the incremental in power generation in every cooling system, then calculate the selling price and cost price to see the value of benefit (inflows) for each system. Then the payback period, which is very important for the investor, we should know the total costs of the investment and appalling listed formula.

$$Paypack\ Period = \frac{Costs\ of\ the\ investment}{Annual\ Cash\ Inflows} \quad \textbf{(40)}$$

1. Between Base Case and Evaporative Cooling System Integration

IPG = EC – BC

Total Selling Price = IPG * Domestic selling price ($/ KWH)

Incremental in fuel consumption (IFC) = EC – BC (Litter)

Cost of fuel = (IFC) * Domestic selling price (0.35 $ / Litter)

Operation and Maintenance cost = IPG * 0.0055 ($ / KWH)

Revenue = Selling Price – Cost of Fuel – Operation and Maintenance.

2. Between Base Case and Absorption Cooling System Integration

IPG = AS – BC

Total Selling Price = IPG * Domestic selling price ($/ KWH)

Incremental in fuel consumption (IFC) = AS – BC (Litter)

Cost of fuel = (IFC) * Domestic selling price (0.35 $ / Litter)

Operation and Maintenance cost = IPG * 0.0055 ($ / KWH)

Revenue = Selling Price – Cost of Fuel – Operation and Maintenance.

CHAPTER SIX

CONCLUSIONS AND FUTURE WORK

6.1 Conclusion

Improving resource utilization is becoming increasingly important today, as the request for high quality power sources continues to increase. On the other hand variation of the ambient conditions showed that gas turbine power generator set loss big amount from its design performance at ISO conditions. Fig. 6.2 shows that for every 1C° increased in air inlet temperature, GT loss 1.2 %from the power output. The base burden power yield of an ignition turbine relies on upon air mass stream rate. As the temperature of air entering a burning turbine expands, air thickness diminishes creating the air mass stream rate to diminish; thus, high encompassing temperatures cause turbine power yield diminish.

There are different ways to enhance the execution of gas turbine power plants that work under hot encompassing temperatures a long way from the ISO models. One demonstrated methodology is to lessen the compressor consumption air temperature by introducing an outer cooling framework. In the present study, two distinctive cooling strategies for decreasing air consumption temperature in gas-turbine-based force plants have been inspected, media evaporative cooling and retention cooling. General Electric gas turbine model MS9001E at Duhok Gas Power Station owned and operated by Mass Holding Group, Duhok, Iraq, have been chosen as experiment for the investigation of the benefits from incorporation of the distinctive admission air-cooling techniques. The calculation were performed on a yearly premise of operation and the time-shifting climatic conditions were checked. Fig. 6.6 illustrate average monthly ambient temperature distribution during operation year 2013. The selection of the period for using cooling system was done by choosing the highest monthly temperature for six months which is start from May and end to October.

Fig. 6.1 shows the incremental in power generation which could be occur through adding two different cooling type comparison with the bas-case, also its obvious that electricity demand increased during the Summer days in the mean while air temperature increase also this lead to more losses in power generation. In average evaporative cooling system increased power output by 27.6%, while absorption cooling system increased power output by 29.8 %.

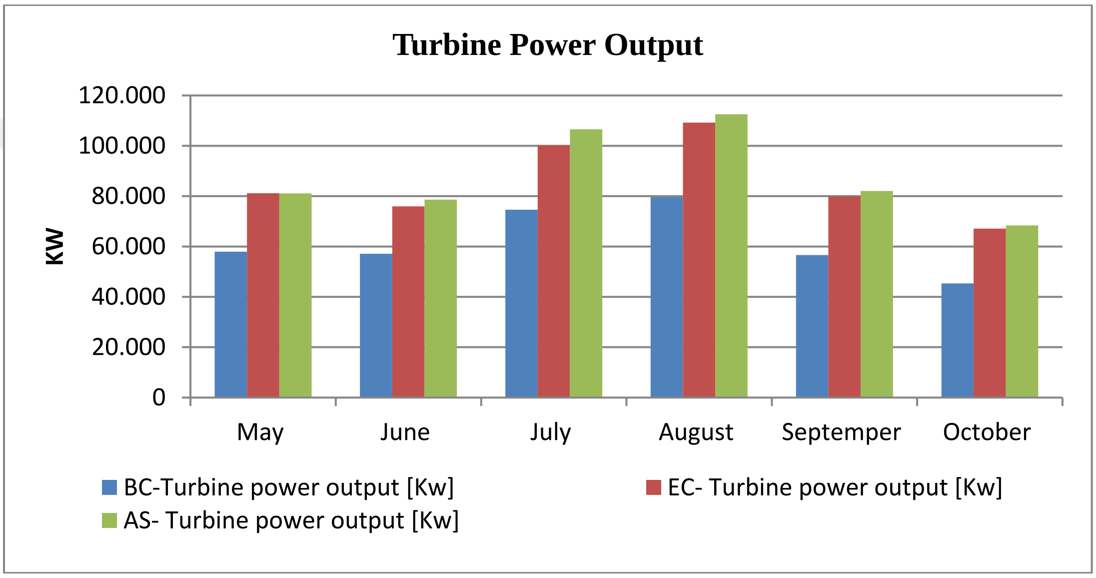

Figure 6.1: Base and incremental in turbine power output during hot time DGPS.

This incremental includes also thermal efficiency of the unit although a part from power generated is use to operate equipment and device of the cooling system. Fig. 6.2 illustrate the improvements in the thermal efficiency of the GT under consideration, the evaporative cooling brought an augmentation of 26.4 %and the absorption chiller gain an 25.5%.

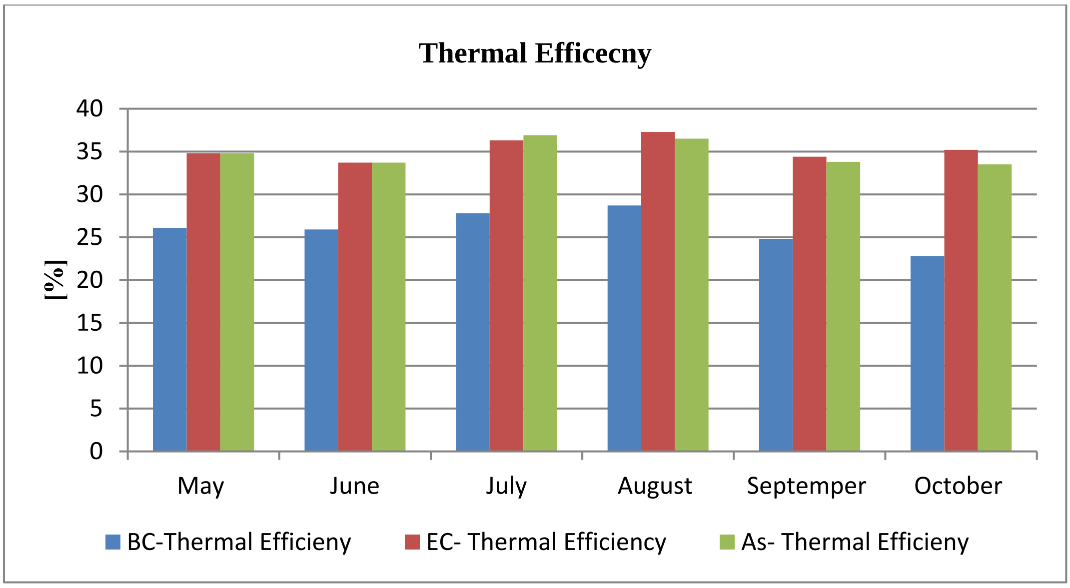

Figure 6.2: Base and incremental in thermal efficiency output during hot time DGPS.

Results demonstrated that both techniques enhance the force yield and warm productivity when contrasted and base-case. By the by, the evaporative cooling strategy was constrained by the encompassing wet-knob temperature, speaking to a reasonable arrangement at low surrounding relative stickiness channel conditions. Then again, the ingestion chiller achieved a bigger temperature drop at various surrounding conditions and gives a full control of the compressor bay conditions paying little mind to encompassing conditions; be that as it may, it requests an entirely vast operational force. Subsequently, if the fumes gas vitality is accessible, this technique speaks to a superior alternative once it can be used free of the encompassing relative stickiness level.

Note that any cooling framework require extra segments. For instance, the assimilation chiller needs a warmth recuperation gadget to use the gas fumes vitality. In any case, these additional parts present a sub-par cost when contrasted and an expansive basic cycle gas turbine motor. In this manner, the best cooling elective must check a few variables as gas turbine parameters, power plant presented limit, load operation sort, site region, climatic conditions, pined for cooling potential, and budgetary feasibility.

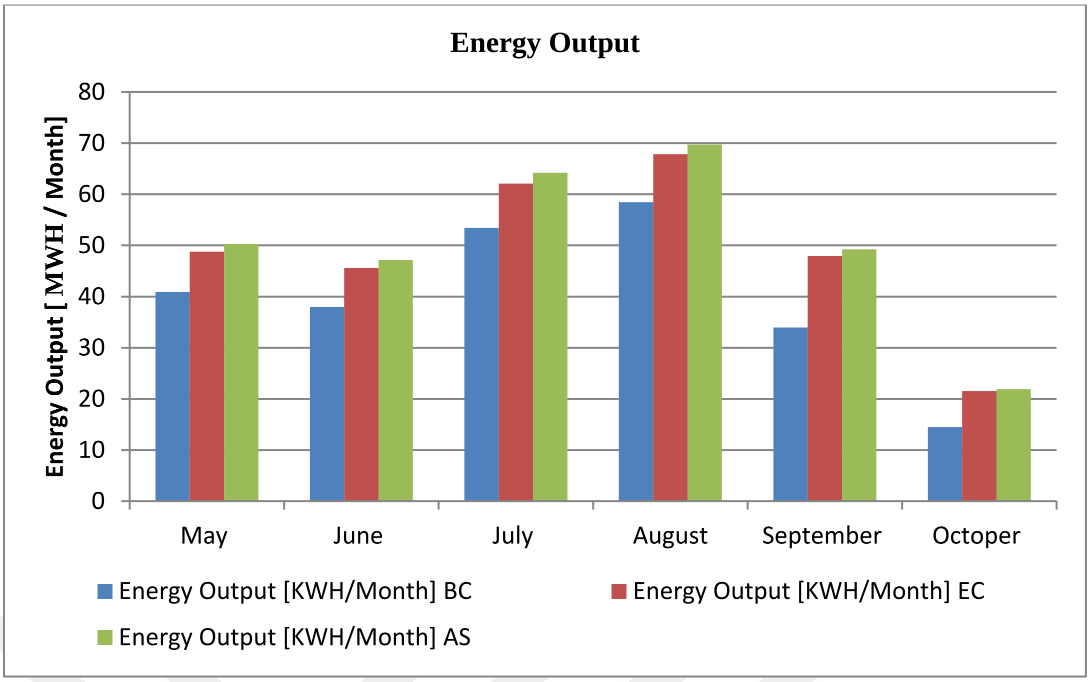

Figure 6.3: Base and incremental in the energy output during hot time DGPS.

The financial aspects from joining of the distinctive cooling were figured and looked at. The outcomes have exhibited that the most elevated incremental power produced is acknowledged by absorption intake air-cooling system as shown in Fig.6.3. Another factor has a great effect from the economical side, which is the increase in fuel consumption to keep turbine inlet temperature constant after adding cooling system, this mean that additional cost for the additional quantity of fuel must be considerate. Fig. 6.4 clarify the increasing of the fuel for each cooling system compared with the base case.

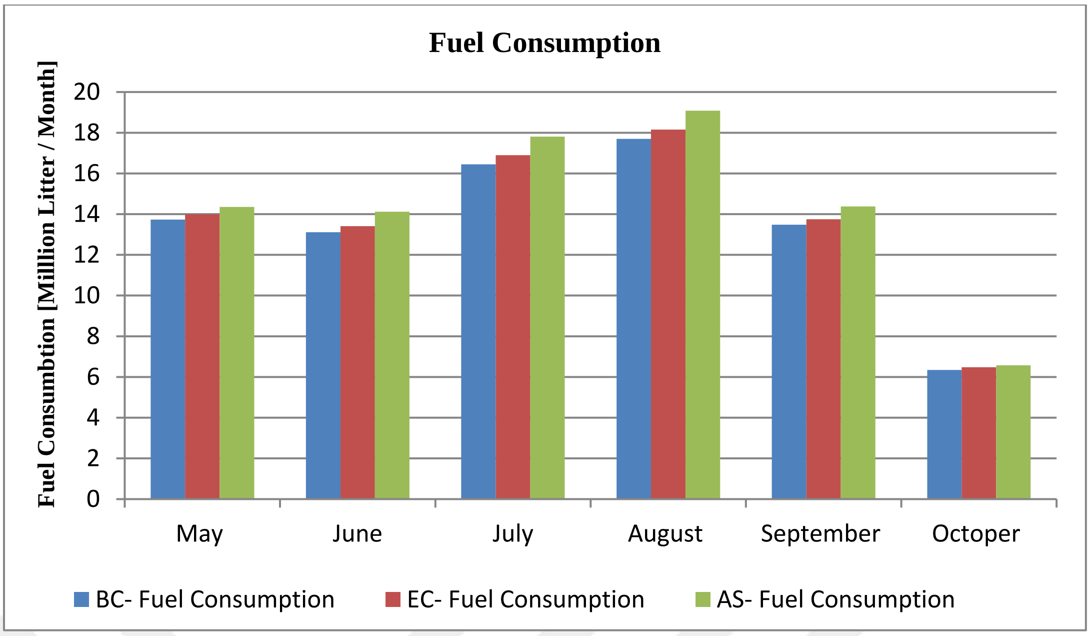

Figure 6.4: Increasing in fuel consumption during hot time DGPS

As far as the financial execution of the venture, the evaporative cooler has the most reduced aggregate expense of incremental power era and the least PB, on the opposite side, ingestion chiller cooling innovation has favorable circumstances of extensive variety of conformities and cooling degrees, which is appropriate for different sorts of gas turbine power plants. Table 6.1 shows total electricity generation costs after and before mix of the gas turbine channel air-cooling framework with the incremental power era (MWh) because of incorporation of the cooling frameworks. The evaporative cooler and the retention chiller framework may both be chosen for boosting the execution of the gas-turbine-based force plants, contingent upon the overarching prerequisites of the plant administrator.

Table 6.1: Electricity generation costs and benefit before and after integration cooling system.

Months	Condition	Energy Output [Kwh/Month]	Fuel Consumption [Litter / Month]	Incremental in power generation (IPG) between BC& EC and BC&AS [KWH]	Selling Price [$]	Incremental in fuel consumption (IFC) between BC & EC and BC&AS [Litter]	Cost of fuel [$]	Operation & Maintenance cost [$]	Benefit [$]
May	Base Case (BC)	40,920,000	13,734,424						
	Evaporative cooling system (EC)	48,803,000	13,898,087	7,883,000	244,373	163,663	57,282	43,357	143,734
	Absorption cooling system (AS)	50,276,000	14,350,987	9,356,000	290,036	616,563	215,797	51,458	22,781
June	Base Case (BC)	38,000,000	13,111,398						
	Evaporative cooling system (EC)	45,560,000	13,409,145	7,560,000	234,360	297,747	104,211	41,580	88,569
	Absorption cooling system (AS)	47,159,000	14,120,179	9,159,000	283,929	1,008,781	353,073	50,375	-119,519
July	Base Case (BC)	53,415,000	16,449,032						
	Evaporative cooling system (EC)	62,098,000	16,893,400	8,683,000	269,173	444,368	155,529	47,757	65,888
	Absorption cooling system (AS)	64,229,000	17,805,199	10,814,000	335,234	1,356,167	474,658	59,477	-198,901
August	Base Case (BC)	58,450,000	17,699,022						
	Evaporative cooling system (EC)	67,837,000	18,151,297	9,387,000	290,997	452,275	158,296	51,629	81,072
	Absorption cooling system (AS)	69,756,000	19,082,517	11,306,000	350,486	1,383,495	484,223	62,183	-195,920
September	Base Case (BC)	33,966,000	13,481,047						
	Evaporative cooling system (EC)	47,919,000	13,747,350	13,953,000	432,543	266,303	93,206	76,742	262,595
	Absorption cooling system (AS)	49,288,000	14,381,529	15,322,000	474,982	900,482	315,169	84,271	75,542
October	Base Case (BC)	14,505,000	6,348,000						
	Evaporative cooling system (EC)	21504000	6,480,000	6,999,000	216,969	132,000	46,200	38,495	132,275
	Absorption cooling system (AS)	21870000	6575974	7,365,000	228,315	227,974	79,791	40,508	108,017

6.2 Future Works

The suggestion for future work is to test more models and different cooling system, especially unprecedented inlet air cooling methods using turbo-expander, which done by reduce the pressure of the natural gas from high pressure to low pressure by development sort ball valves, the temperature lessen as well and this makes extensive cooling limit which can be utilized to enhance execution of the gas turbine.

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