

UNIVERSITY OF BAHRAIN

College of Engineering



**Department of Mechanical
Engineering**

Techno-Economic Analysis of Combined Cycle Gas Turbine Air Intake Cooling Methods for Bahrain Operating Conditions

A Thesis submitted in Partial Fulfilment of the Requirements for the
M.Sc. Degree in Engineering Management

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Abstract

One of the main problems of the combined cycle gas turbine plants is the sensitivity to the air intake temperature. The increase in the intake temperature reduces the plant's output and efficiency. In Bahrain, the air temperature reaches around 48 °C during summer, negatively impacting the power plants' performance.

Based on technical and economic analysis, this thesis investigated the feasibility of adding different air intake cooling systems on combined cycle gas turbine plants in Bahrain. Fogging and evaporative cooler systems have been selected to represent the evaporative cooling method. In addition, mechanical and absorption chillers have been chosen to represent the refrigeration cooling method.

The selected combined cycle gas turbine plant for this work contains two gas turbines combined with a steam turbine. The total design output of the selected plant is 491 MW. The plant has been modeled and simulated in Ebsilon professional software based on the collected manufacturer data and Bahrain's climate data for 2022.

The technical analysis results have shown that the waste heat absorption chiller, internal steam absorption chiller, mechanical chiller, fogging system, and evaporative cooler increased the annual plant generation by 6.85%, 6.23%, 5.89%, 5.16%, and 4.33%, respectively. In addition, the fogging system, waste heat absorption chiller, and evaporative cooler increased the annual plant efficiency by 0.16%, 0.12%, and 0.10%, respectively. On the other hand, the mechanical and internal steam absorption chillers reduce the annual plant efficiency by 0.78 % and 0.46%, respectively.

The economic analysis results have shown that net present values of the waste heat absorption chiller, fogging system, internal steam absorption chiller, mechanical chiller, and evaporative cooler were \$ 124 M, \$ 113 M, \$ 101 M, \$ 95 M, and \$ 85 M respectively.

This research has concluded that the waste heat absorption chiller is the best alternative to cool the air intake of the gas turbines operated on a combined cycle arrangement in Bahrain, provided that the plant has a reasonable waste heat source. Otherwise, the fogging system should be prioritized, followed by the internal steam absorption chiller, mechanical chiller, and evaporative cooler, respectively.

Dedication

I want to dedicate this work to my parents, who supported me on all levels.

I also want to dedicate this work to my wife, who made many sacrifices throughout the study period.

Acknowledgment

I want to thank all those who contributed to this work, which otherwise would not have been achieved.

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Nomenclature

A	[m ²]	Area
B	[m ³ /s]	Blow Down Rate
C_f	[\$]	Annual Additional Fuel Cost
C_i	[\$]	Insallation Cost
$C_{O\&M}$	[\$]	Annual Operation and Maintenance Cost
C_w	[\$]	Annual water cost
$C.F$	[-]	Correction Factor
C_p	[kJ/kg.K]	Specific Heat Capacity at Constant Pressure
EX	[-]	Mass Flow Rate Exponent
LHV	[kJ/kg]	Lower Heating Value
$\dot{m}$	[kg/s]	Mass Flow Rate
N	[Year]	Number of Years
P	[MW]	Power Output
$\dot{Q}$	[kW]	Heat Transfere Rate
R	[m ³ /s]	Evaporation Rate
T_{DB}	[°C]	Dry Bulb Temperature
T_m	[°C]	Mean Temperature
T_{WB}	[°C]	Wet bulb Temperature
TET	[°C]	Turbine Exhaust Temperature

h	[kJ/kg]	Specific Enthalpy
HR	[kJ/kWh]	Heat Rate
P	[MW]	Power Output
R	[m ³ /s]	Evaporation Rate
U	[kW/m ² K]	Heat Transfer Coefficient

Special characters

η	[%]	Efficiency
ω	[kg_{water}/kg_{air}]	Specific Humidity
ρ	[kg/m ³]	Density

Subscripts

C	Cooling
c	Cold
CC	Combined Cycle
CL	Cooling Load
cs	Cold side
d	Design
da	Dry air
E	Expected
ext	Extracted
f	Fuel
g	Gas

<i>Gen</i>	Generator
<i>h</i>	Hot
<i>hs</i>	Hot side
<i>isen</i>	Isentropic
<i>mech</i>	Mechanical
<i>OC</i>	Open Cycle
<i>od</i>	Off-Design
<i>Re</i>	Reference
<i>s</i>	Steam
<i>w</i>	Water

Abbreviation

A.T	[°C]	Ambient Temperature
ACC	[-]	Air Cooled Condenser
C.I.RH	[°C]	Compressor Inlet Relative Humidity
C.I.T	[°C]	Compressor Inlet Temperature
CCGT	[-]	Combined Cycle Gas Turbine
CEP	[-]	Condensate Extraction Pump
COP	[-]	Coefficient of Performance
ECO	[-]	Economizer
EPD	[-]	Exhaust Pressure Drop
EVA	[-]	Evaporator
FWP	[-]	Feed Water Pump
GT	[-]	Gas Turbine
HHV	[kJ/kg]	Higher Heating Value
HL	[-]	Heat Loss
HP	[-]	High Pressure
HR	[kJ/kWh]	Heat Rate
HRSG	[-]	Heat Recovery Steam Generator
IPD	[-]	Inlet Pressure Drop
IRR	[%]	Internal Rate of Return
LHV	[kJ/kg]	Lower Heating Value

LMTD	[-]	Log Mean Temperature Difference
LP	[-]	Low Pressure
NPV	[\$]	Net Present Value
PB	[Years]	Pay Back Period
RH	[%]	Relative Humidity
SH	[-]	Super Heater
ST	[-]	Steam Turbine
UTTD	[-]	Upper Terminal Temperature Difference

Chapter One

Introduction

1.1 Introduction

1.2 Research Problem

1.3 Research Questions

1.4 Research Objectives

1.5 Research Significance

1.6 Research Setting

1.7 Research Terminology

Chapter one: Introduction

1.1 Introduction

Nowadays, the availability of electricity is considered to be one of the most important elements of life. Electricity needs a primary source of energy such as fossil, renewable, or nuclear to be generated. As per the British Petroleum statistics report, fossil fuels have been used to generate 61.4% of the entire world's electricity in 2021 (BP, 2022).

To generate electricity from fossil fuels, a machine should transform the fuel energy into a work that can rotate an electrical generator. The gas and steam turbines are the most famous machines used for that purpose. The gas turbine (GT) is an engine that transforms the natural gas energy to work in the GT shaft, which can be used to rotate the generator's rotor. It works under a thermo dynamic cycle called the Brayton cycle. The GT process starts with compressing the ambient air with an axial flow compressor. Then, the fuel is mixed with the pressurized air inside a combustion chamber to increase the mixture temperature. Finally, the turbine expands the mixture to convert its thermal energy into work.

The steam turbine (ST) utilizes the energy in the pressurized steam to produce work. The ST works under a thermo-dynamic cycle called the Rankine cycle, which uses water and steam as working fluids. The process starts by pumping water from the feed water tank to a boiler. Then, the boiler converts the supplied water to steam by using burners. Next, the steam energy is converted to work by a turbine through an expansion process. Finally, the exhaust steam is cooled to convert it to a liquid form to be pumped back into the feed water tank.

The GT has two different operation arrangements. Firstly, the open cycle, where the hot gas is exhausted into the atmosphere after the turbine.

Secondly is the combined cycle, where the gas heat after the turbine is utilized for steam generation before dumping into the atmosphere. Therefore, running under a combined cycle arrangement is considered the best operating strategy for maximizing power generation efficiency (Suaal, 2015).

Various solutions have been made to enhance the combined cycle gas turbine plant's performance. Most of the improvements are related to the GT side, which significantly influences the overall plant performance (Jones & Jacobs, 2000). One of the critical gas turbine drawbacks is the sensitivity to the ambient temperature. Generally, for every one-degree Celsius increase in the compressor intake temperature, the output of the open and combined cycles is decreased by 0.7 % and 0.45%, respectively (Wan et al., 2021). This phenomenon occurs because the air density decreases with the increase of its temperature. Therefore, less air is entered into GT, which reduces the output. In addition, the GT compressor requires more work to compress the same air volume at a higher inlet temperature. Therefore, the increase in intake temperature reduces GT output and efficiency, which affects the combined cycle performance (Ibrahim & Rahman, 2013).

An air intake cooling system should be used to reduce the impact of the high intake temperature. Evaporative and refrigeration are the two main cooling methods used for that purpose (Yang et al., 2009). The wet bulb temperature of the air restricts the amount of cooling by an evaporative cooling process. However, refrigeration cooling can cool the intake to any desired temperature. The present work aims to select the best technical and economical air intake cooling system for combined cycle gas turbine plants in Bahrain.

1.2 Research Problem

The combined cycle gas turbines (CCGT) plants are widely commissioned nowadays for electricity generation due to their flexibility and high thermal efficiency. Bahrain's government depends on AL EZZEL, HIDD, ALDDUR 1, and ALDURR 2 CCGT plants to generate the required electricity demand. These plants are owned by the government but operated and maintained by private-sector companies.

Any private sector power provider aims to maximize profit, which can be achieved by increasing electricity sales to the national grid. Bahrain's electricity demand increases dramatically during summer (EWA, 2021), which is an opportunity for private sector providers to increase their sales. However, the ambient temperature reaches around 48 °C at the peak summer in Bahrain, which reduces the CCGT plant performance.

The impact of Bahrain's harsh weather on CCGT plants could be reduced if the air intake cooled before the compressor section. Different air intake cooling systems are available for that purpose. The main two categories are evaporative cooling and refrigeration cooling (Deng et al., 2020). The amount of cooling done by an evaporative cooling system depends on the wet bulb temperature. In addition, this method requires a continuous water supply, which could be a limitation nowadays with the world's attention to water conservation. On the other hand, there are no cooling restrictions to cool the air by a system that is based on refrigeration cooling, but it costs much more than evaporative cooling systems (Deng et al., 2020).

A system that can provide more considerable power augmentation for the CCGT is not necessarily the best economical option. The selection of an appropriate system strongly depends on site climate and resource availability.

The present work aims to select the most appropriate intake cooling system for Bahrain's operating conditions based on technical and economic considerations.

1.3 Research Questions

To choose the best technical and economical CCGT air intake cooling technology, many critical questions need to be answered as follows:

- What is the yearly pattern of Bahrain's climate condition?
- What is the effect of Bahrain's climate on CCGT performance?
- What is the impact of cooling the air intake of CCGT?
- What are the gas turbine air intake cooling systems available?
- What are the features, drawbacks, and costs of each cooling system?
- Which cooling system is the best option for CCGT air intake cooling in Bahrain?

1.4 Research Objectives

The objectives of the thesis are as follows:

- To model and validate a CCGT plant.
- To explore the effect of air intake temperature on the selected plant performance.
- To simulate the build model of the CCGT plant concerning Bahrain's yearly climate pattern.
- To determine the impact of different air intake cooling systems on CCGT plants located in Bahrain.
- To choose the best technical and economical CCGT air intake cooling system as per Bahrain's operating condition.
- To give recommendations for all combined cycle power plants operating in Bahrain.

1.5 Research Significance

Currently, CCGT plants are supplying all electricity demand in Bahrain. Therefore, even small changes in their performance could make a difference for Bahrain. By the end of this research, the outcomes will be as follows:

- It would give enough support for the decision-makers in Bahrain to make the best investment in one of the CCGT air intake cooling systems.
- It will reduce the cost of electricity generation in Bahrain by preventing starting more machines to meet the same electricity demand.

1.6 Research Setting

This research will study the technical and economic feasibility of different CCGT air intake cooling methods using Bahrain's weather data for 2022. The results of this research are sensitive to the climate and they are applicable in Bahrain only, or other countries that have similar climates.

1.7 Research Terminologies

- **Gas Turbine (GT):** A machine that transforms the fuel energy to work in the shaft.
- **Steam Turbine (ST):** A machine that transforms the steam energy to work in the shaft
- **Heat Recovery Steam Generator (HRSG):** a device that is used to generate steam by using the heat of hot gas.
- **Combined Cycle Gas Turbine (CCGT):** a power plant that generates power by combining the gas and steam turbine cycles by utilizing the heat of the GT exhaust to produce steam for the steam turbine.

Chapter Two

Literature Review

2.1 Impact of Intake Temperature on CCGT Plants

2.2 Air Intake Cooling Systems for CCGT

2.3 Summary

Chapter Two: Literature Review

2.1 Impact of Intake Temperature on CCGT plants

The GT uses the air as a working fluid. Therefore, any changes in the air parameters affect the machine's performance in terms of power output, efficiency, exhaust temperature, and exhaust flow (Brooks, F. J. 2000).

The effect of the air temperature on the CCGT plants has been studied widely in the open literature. (Ibrahim & Rahman, 2013) Explore the effect of compressor intake temperature on a three-pressure level with reheat and supplementary firing CCGT plant performance. The research model was built and simulated in THERMO FLEX software. The results show that by varying the intake temperature from 0 °C to 50 °C, the plant's output decreases from 880 MW to 820 MW. Furthermore, the plant's efficiency decreased from 55.5% to 52.6%. The researchers conclude that at high compressor intake temperature, the output and efficiency of the plant are reduced due to less inlet mass and high work requirement by the air compressor, which, therefore, affects the overall cycle performance.

(Tiwari et al., 2013) Modeled the CCGT plant of India's national thermal power corporation plant using the laws of thermodynamics. The selected plant involves four GT and two ST, with a total plant output of 817 MW. The research investigated the relationship between ambient temperatures and efficiency. The results show that the plant efficiency decreased by approximately 0.04% per each degree Celsius increase.

(Şen et al., 2018) studied the effect of intake temperature on the performance of CCGT. The researchers collected 14 years of actual operation data from a CCGT plant in Turkey with a range of 8 ° C to 23 ° C. The results show that at a temperature of 8 ° C, the total plant output and efficiency were 227.7 MW and 43.3%, respectively. However, at an

air temperature of 23 ° C, the plant's power output and efficiency were 197.3 MW and 42.7%, respectively. The research concludes that ambient temperature is a vital function in CCGT plant performance and suggests utilizing an air intake cooling technology to reduce the losses.

(Wan & Chen, 2021) studied the degradation of CCGT plant performance in China during high ambient temperatures based on actual operational data. The model of the plant was based on a PG9351FA GT coupled with a D10 ST by a three-pressure levels HRSG. The results show that the CCGT plant output decreased from 433 MW to 335 MW when the air temperature increased from 5 °C to 35 °C. In addition, the same air temperature drop reduced the plant efficiency from 57.28% to 56.3%. The study concludes that plant power output has been reduced because of the decrease in the air and fuel mass flow as the intake temperature increases. In addition, the efficiency has dropped because of the high power consumption by the air compressor.

2.2 Air Intake Cooling Systems for CCGT

Many researchers have studied and investigated the feasibility of intake cooling to reduce the impact of high ambient temperature on CCGT plant performance.

(Hiwa Khaledi et al., 2005) explore and investigate the effect of cooling the intake of CCGT using an absorption chiller. The researchers compared external and internal steam sources to operate the absorption chiller to investigate each source's impact on the CCGT output and efficiency. The researchers selected a GE 9171 E frame gas turbine linked to two pressure levels HRSG, which had been modeled based on the thermodynamics laws. The results showed that the CCGT output increased by around 17%, 9%, and 8% using an external source absorption chiller, internal source two-

stage absorption chiller, and internal source one-stage absorption chiller, respectively, at a temperature of 45 °C. In addition, 8% and 6% output reductions have been noted using an internal one-stage absorption chiller and an internal two-stage absorption chiller, respectively. However, the external source absorption chiller does not affect the cycle efficiency. It has been concluded that using an internal steam source reduces combined cycle augmented power and cycle efficiency due to the extracted steam from the HRSG. On the other hand, using an external steam source enhances the plant's output without affecting its efficiency.

(BOONNASA et al., 2006) They have provided a techno-economical investigation on adding an internal steam source absorption chiller to a CCGT plant in Thailand. The plant involves two GT and one ST with An output of 337 MW. Results have shown that the absorption chiller enhanced the output by 6.24% with a payback period of 3.8 years and \$ 19.44 M as a net present amount of the project. However, the research concludes that extracting steam from an internal source limits the power output and efficiency improvements. Therefore, better results could be obtained using another source, like waste heat.

(Saeed et al., 2011) studied the impact of using different inlet air cooling systems on a CCGT plant in Kazeroun, Iran, based on technical and economic considerations. The air inlet fogging system, evaporative cooler, water-cooled mechanical chiller, air-cooled mechanical chiller, and absorption chiller have been adopted for this research. The selected CCGT plant contains two GT and one ST with a total output of 478 MW. The thermoflow simulation package was used to model and simulate the selected plant. The results have shown that the water-cooled mechanical chiller is the best system that could be used. The NPV of the selected system was \$ 12 M with a PB period of 3 years.

(Njoku, 2018) investigated the feasibility of adding a waste heat absorption chiller on the intake air stream of the GT and the ACC on a CCGT plant in Nigeria. The researcher uses the heat available after the HRSG to drive the chiller. The results have shown that by reducing the GT intake air temperature to 15 °C, The output of the plant was enhanced by 48 MW. Moreover, by fixing the air temperature that entered the ACC at 29 °C, the net increase of ST output is only 1.4 MW. In addition, the results have shown that using the waste heat after the HRSG to drive the absorption chiller reduces the flue gas temperature after the HRSG to 84 °C. The economic study shows that the total plant annual revenues and NPV increased by 7.7% and 17%, respectively. The emissions analysis has shown that the total Nox and CO₂ emissions decreased by 65% and 5.6%, respectively. The researcher concluded that using a waste heat absorption chiller can improve the CCGT plant from technical, economic, and environmental perspectives.

(Elberry et al., 2018) investigated the impact of using an inlet single-stage waste heat absorption chiller on the GT inlet that operated under the CC arrangement. Nubaria combined cycle plant in Egypt has been modeled for this research. The selected absorption chiller uses lithium bromide as a refrigerant and the exhaust gases after the HRSG as a heat source. The model has been built based on thermodynamics equations and then simulated in EES. As per the results and by fixing the HRSG exhaust temperature at 122 °C and the surrounding air temperature at 45 °C, the output increased from 620MW to 650MW (increased by 4.84%), and thermal efficiency increased from 55% to 55.125% (increased by 0.23%). Moreover, by utilizing the selected plant's weather data for 2015, the plant generation shows an improvement of around 11%.

(Deng et al., 2020) provided a comprehensive review on using different air intake cooling systems for CCGT. The researchers investigated refrigeration and evaporative cooling systems, including their working principles, advantages, drawbacks, and costs. By reviewing a massive number of results, the results have shown that the fogging system could improve CCGT performance by up to 17% compared to 4% only if evaporative coolers were used. On the other hand, the mechanical chillers could improve the CCGT performance by up to 13.6% compared to a 23% improvement by using absorption chillers. Research concludes that absorption chillers are the best choice if the CCGT performance improvement is the main priority in selection. In addition, the researchers suggest some factors need to be considered to make an optimized selection: the climate profile, the initial cost, the augmented power generated, additional gas and water costs, and the impact on the environment.

2.3 Summary

The literature review has shown the effect of compressor intake temperature on the CCGT plant, which occurs mainly due to the reduction of air density. The most famous solution is to install an air intake cooling system on the GT intake system. Most systems used were fogging, evaporative coolers, and mechanical and absorption chillers. The drawback of evaporative cooling systems is the cooling restriction due to the relative humidity, which makes them unsuitable for countries with humid climates. However, the evaporative cooling systems could be attractive due to their lower cost. On the other hand, refrigeration cooling systems can augment the plant output more than evaporative systems because they do not have any cooling restrictions. However, installing and maintaining refrigeration systems is much more expensive compared to evaporative cooling systems.

To sum up, the researchers agreed that the selection of the cooling technology depends on several factors, such as the plant location, available resources, and cost.

Chapter Three

Methodology

- 3.1 Introduction
- 3.2 Gas Turbine Modeling
- 3.3 HRSG Modeling
- 3.4 Steam Turbine Modeling
- 3.5 Deaerator Modeling
- 3.6 Feed Water Pre-Heater Modeling
- 3.7 Air Cooled Condenser Modeling
- 3.8 Water Pumps Modeling
- 3.9 Combined Cycle Performance
- 3.10 Gas Turbine Inlet Air Cooling Systems
- 3.11 The Selection Process

Chapter 3: Methodology

3.1 Introduction

This research explores and investigates the feasibility of different air intake cooling systems to cool the inlet air for CCGT plants based on technical and economical analysis for Bahrain operating conditions.

In the technical analysis, a model of the selected CCGT plant will be built first to simulate the base plant behavior under Bahrain Climate data for 2022. Then, the model will be simulated after adding the selected cooling systems under the same climate data.

Inlet fogging and evaporative coolers have been selected to represent the evaporative cooling method. Moreover, the mechanical, internal steam absorption chiller and waste heat absorption chillers have been selected to represent the refrigeration cooling method. Finally, the technical phase will end by determining each cooling system's annual additional energy generation, gas consumption, and water consumption.

The economic analysis will determine each cooling system's initial investment cost, annual cost, annual revenues, and net cash flow to determine different profitability measures.

The modeling procedure of the selected CCGT plant, cooling systems, and the selection process among alternatives will be discussed in this chapter.

3.1.1 The Selected CCGT Plant

The selected plant for this thesis is a CCGT plant that contains two identical GT, two identical HRSG, and one ST. The total designed output of the CCGT plant is 491 MW. Figure 1 shows the selected CCGT plant layout.

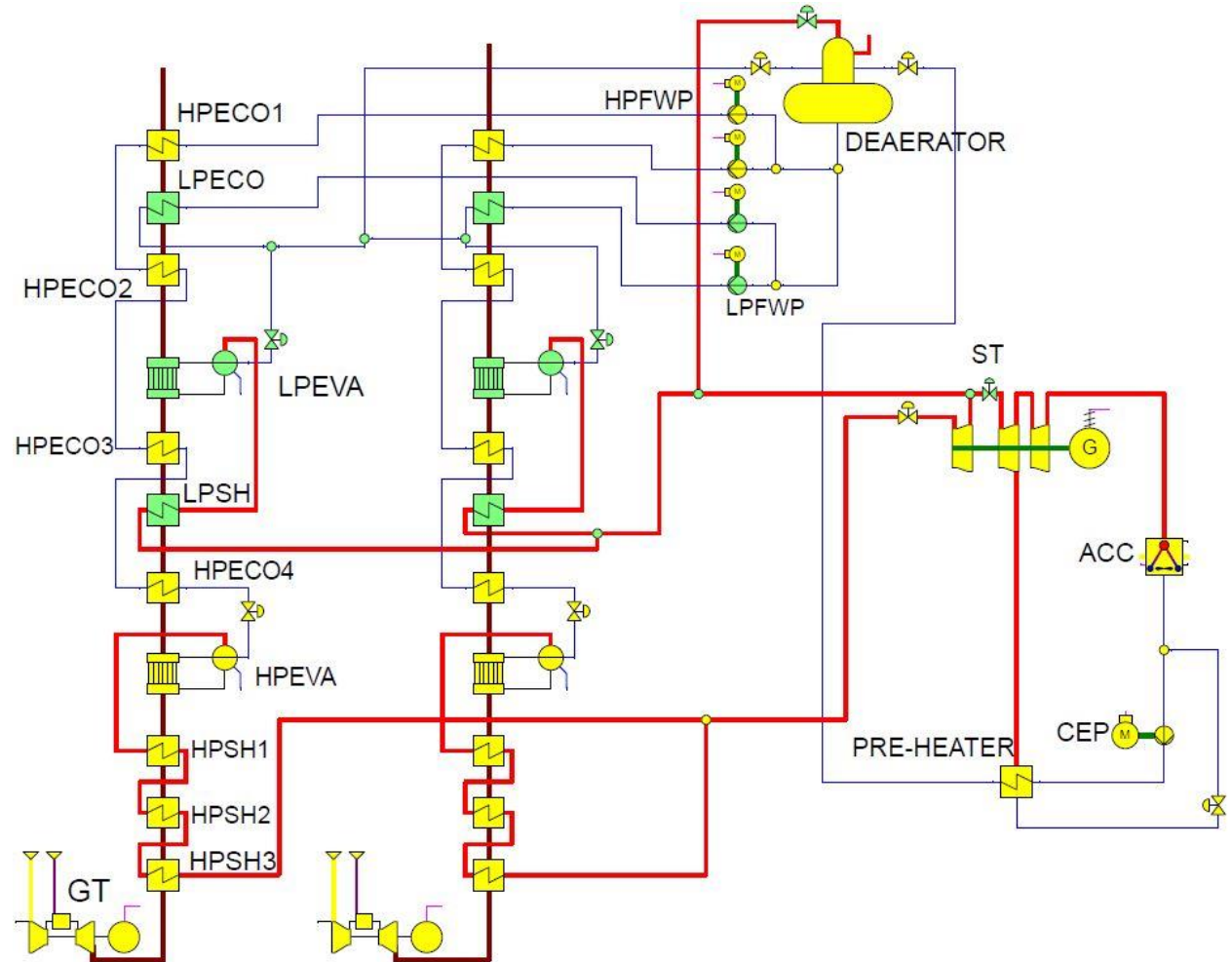


Fig. 1: The selected CCGT plant layout In Ebsilon

3.1.2 CCGT Plant Process description

The process starts in the GT, where the compressor compresses the air. The high-pressure and temperature air that been produced is mixed with the fuel inside the combustion chamber to be ignited. The combustion process product contains high energy, which will be converted to work by the turbine. Finally, the remaining work in the shaft is used to drive the electrical generator.

The gas after the turbine is delivered to the HRSG to generate steam for the ST cycle. The water is transferred from the deaerator to the HRSG by high and low-pressure feed water pumps. Each water stream is passed first through an economizer to increase the feed water temperature to the design value. Then, the water enters the evaporator to produce steam. Finally, the saturated steam in the evaporator entered a superheater, where sensible heat was added to the steam to increase its temperature to the required design outlet temperature.

The produced steam from each HRSG is joined in one header for the HP steam and another for the LP steam. The HP steam is utilized fully in the ST. The deaerator extracts some LP steam while the remaining is admitted to the ST. The ST transforms the thermal energy of the steam to work in the shaft to drive the electrical generator. The ACC cools the steam after the ST to change its phase to liquid. The condensate extraction pump (CEP) transfers the water to the deaerator. The water is passed through a preheater to increase its temperature before it reaches the deaerator. The preheater extracts steam from the ST, which will be condensed and fed into the cycle.

3.1.3 Simulation package

The selected plant has been modeled and simulated using Ebsilon professional software. This simulation package specializes in designing, simulating, and optimizing power plants and thermodynamic cycles. It has been continuously modified and tuned for the last 30 years, which makes it one of the most famous and influential packages used for this purpose (STEAG EBSILON, n.d.). Many researchers have used Ebsilon as an accurate performance estimation tool for different power plant configurations (Egware & Obanor, 2021). For example, (Aminov et al., 2016) used Ebsilon Professional to model a CCGT plant with different

steam pressure levels. The researcher aimed to investigate the performance of CCGT technology for fuel savings, CO₂, and Nox emissions reduction.

(Egware & Obanor, 2021) Modeled an actual gas turbine in Ebsilon professional to study the machine's behavior under different air intake temperatures. The model results have been validated with actual operating data that only gives an error ranging between -3.24% to 2.66%.

(Żyrkowski & Żymełka, 2019) Uses Ebsilon to investigate the loading behavior of a 225 Mw coal-fired plant. The results of the model have been validated using operational plant data.

(Matjanov, 2020) investigate the feasibility of using an intake absorption chiller for the GT to increase machine efficiency. The researchers utilized Ebsilon to build the plant model.

To sum up, many researchers use Ebsilon to model and simulate power plants and thermodynamic cycles. This verifies that this package has been used widely for research purposes in the power and energy field.

3.2 Gas Turbine Modeling

Two identical gas turbines were modeled in Ebsilon using the gas turbine Base load correction curves collected from the manufacturer. The modeling of gas turbines using the correction curves has been used widely in the open literature (Byeong Seon Choi et al., 2016), (Lee et al., 2017) and (Plis & Henryk Rusinowski, 2016).

The GT correction curves are used to determine the off-design behavior of the machine in case any of the reference operating conditions have been changed. The reference operating conditions are the parameters stated by the designer at which the machine can be at the guaranteed design performance. Tables 1 and 2 show the reference operating conditions and

the guaranteed design performance of the selected gas turbine, respectively.

Table (1): The Reference Operating Conditions for the Selected GT

Parameter	Value
Compressor inlet temperature	15 °C
Ambient pressure	1.013 Bar
Relative Humidity	60%
Inlet pressure drop	0 Bar
Exhaust pressure drop	0 Bar
Fuel LHV	34263.02 kJ/kg
Frequency	50 H.z
Injection Flow	0 kg/s
Relative Power	100 % (Base load)

Table (2): The Selected Gas Turbine Performance Ratings at Reference Operating Conditions

Parameter	Value
Power output	185.95 MW
Heat rate	9506 kJ/kWh
Exhaust flow	552.5 kg/s
Exhaust temperature	501 °C

3.2.1 The used GT correction curves for modeling

The following section will discuss all base load correction curves used to model the selected gas turbines in Ebsilon.

3.2.1.1 Inlet temperature correction curves

As discussed, the gas turbine machines operate at a fixed volumetric flow rate at base load. When the intake air temperature increases, its density decreases, which reduces the mass of the intake air and, therefore, reduces GT output. In addition, the compressor work increases at high intake temperatures, causing a higher machine heat rate (Tiwari et al., 2013) (Farouk et al., 2013). Figure 2 shows the GT power output correction curve for inlet temperature. The curve shows that if the compressor intake increases from 15 °C to 50 °C, the GT power output will reduce by 20 %. Moreover, Figure 3 shows the GT heat rate correction curve for inlet temperature. The curve shows that if the compressor intake increases from 15 °C to 50 °C, the GT heat rate will increase by around 7%.

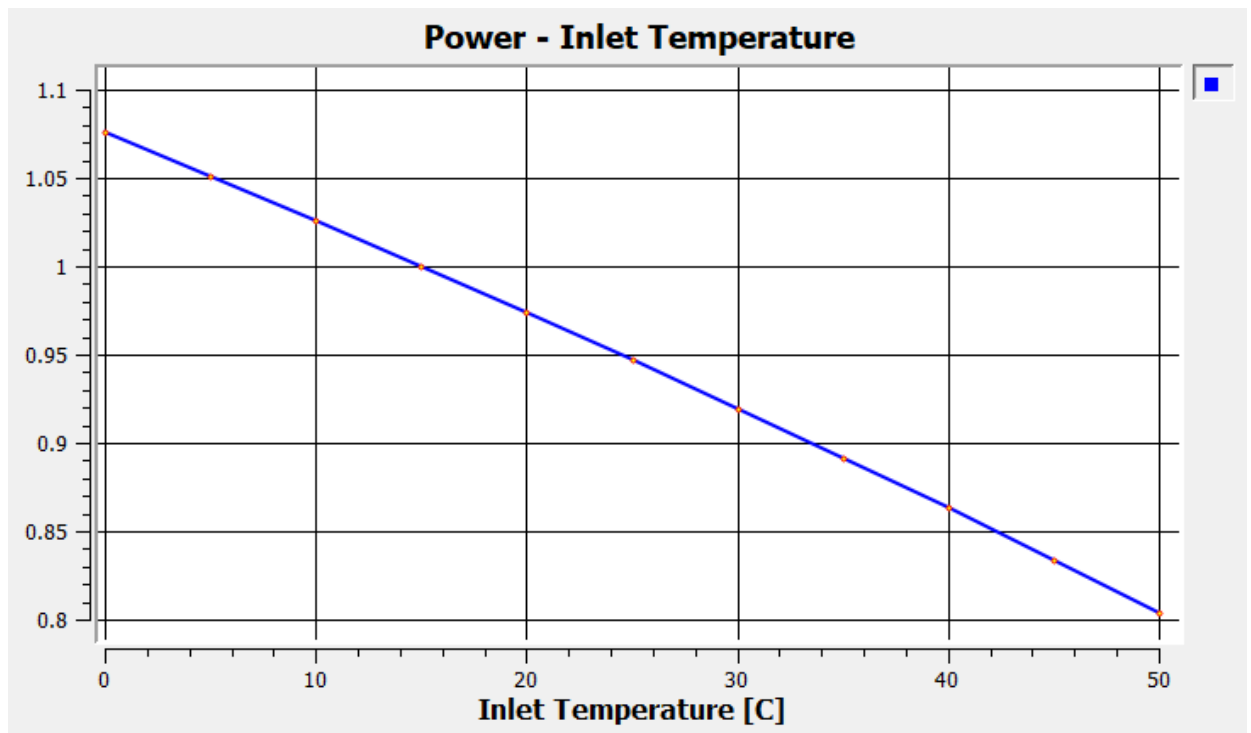


Fig. 2: GT power output correction curve for inlet air temperature

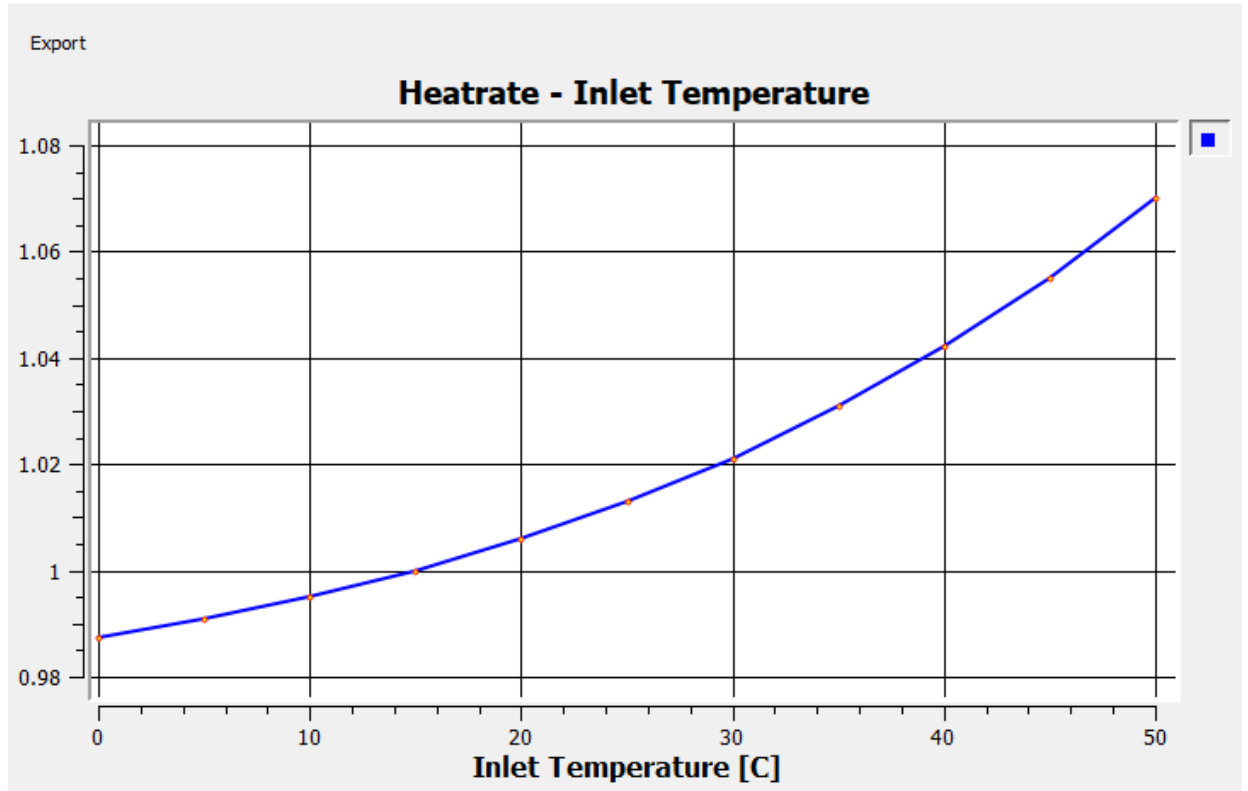


Fig. 3: GT heat rate correction curve for inlet air temperature

Since this research is based on a combined cycle configuration, the effect of the intake temperature on the exhaust condition of GT should be studied. Figure 4 shows the GT exhaust temperature correction curve for inlet temperature. The curve shows that turbine exhaust temperature increases with the increase of compressor intake temperature. This occurred because the exhaust temperature is inversely proportional to the pressure ratio, and the pressure ratio is inversely proportional to the intake temperature. Therefore, as the compressor intake increases, the pressure ratio decreases, and the exhaust temperature increases due to the decrease in the turbine work (Scott et al., 1995). Figure 5 shows the exhaust mass flow correction curve for inlet temperature. As per the curve, the leaving mass of the gas reduces by 13% from 15 °C to 50 °C.

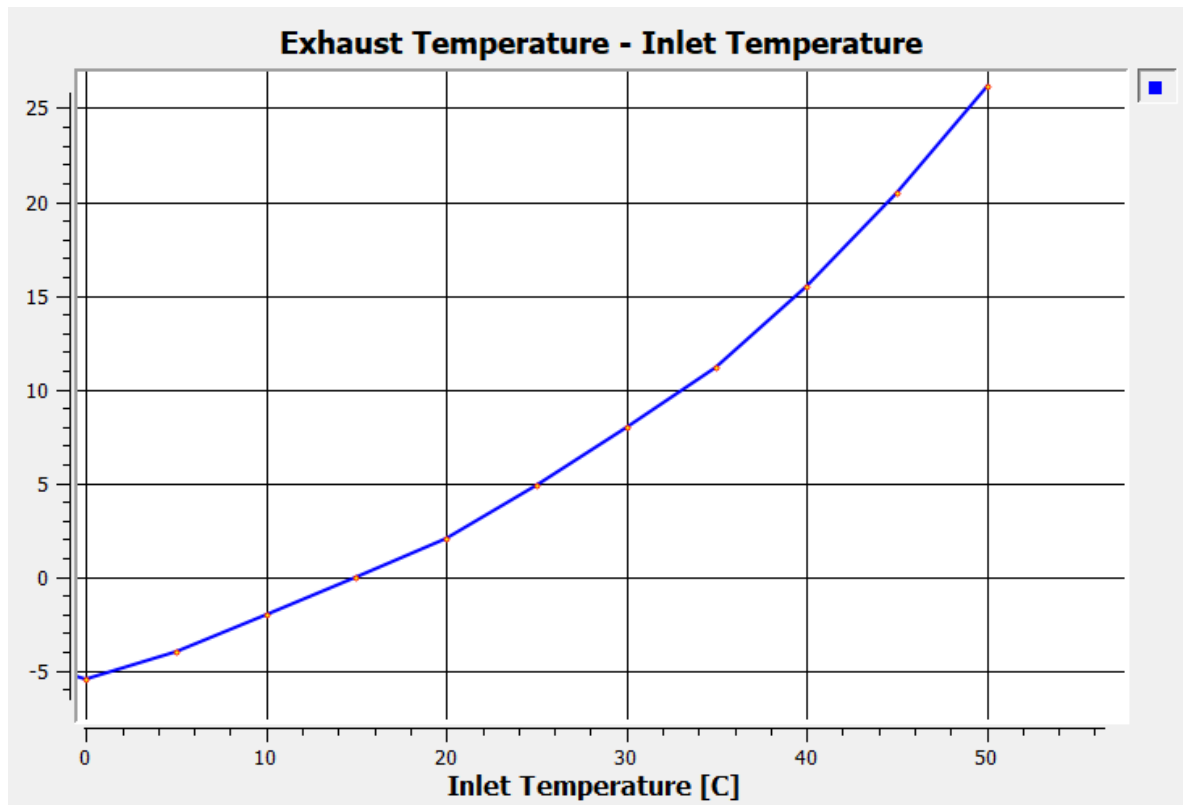


Fig. 4: GT exhaust temperature correction curve for inlet air temperature

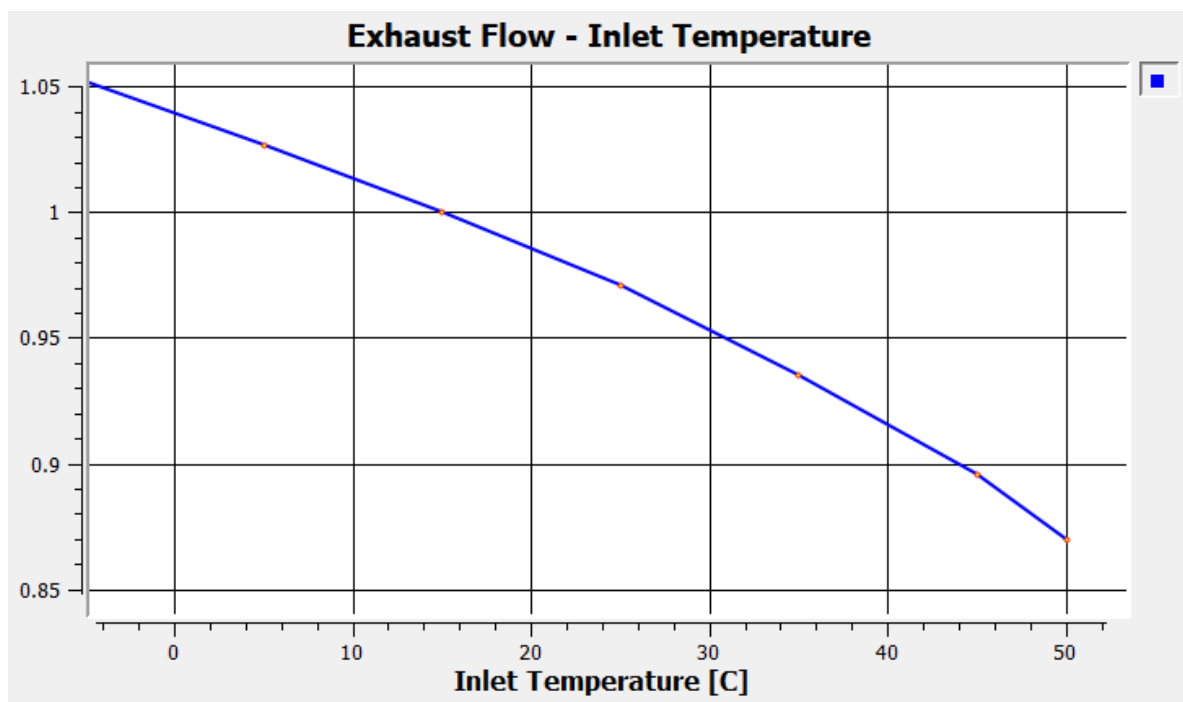


Fig. 5: GT exhaust mass flow correction curve for inlet air temperature

3.2.1.2 Inlet relative humidity correction curves

In general, the dry air density is higher than wet air because the dry air molecular weight (28 g/mol) is heavier than the water (18 g/mol) (Brooks et al., 2000). Therefore, the air density decreases as the air becomes wetter due to the molecular weight differences.

As discussed, the air density affects the air mass flow that enters the GT. When the air density falls, the inlet air mass flow to the compressor falls, which affects the output and heat rate of GT.

The selected turbine for the presented work has a fixed turbine inlet temperature (TIT). Therefore, the control system will always try to keep it at the design value.

The air temperature after the compression process reduces if the inlet RH increases. As a result, the TIT temperature will drop. GT controller increases the inlet fuel mass flow to maintain the TIT at the design value, which leads to a higher machine output and heat rate (Kurz & Brun, 2016). Figures 6 and 7 show the GT output and heat rate correction curves for inlet relative humidity, respectively.

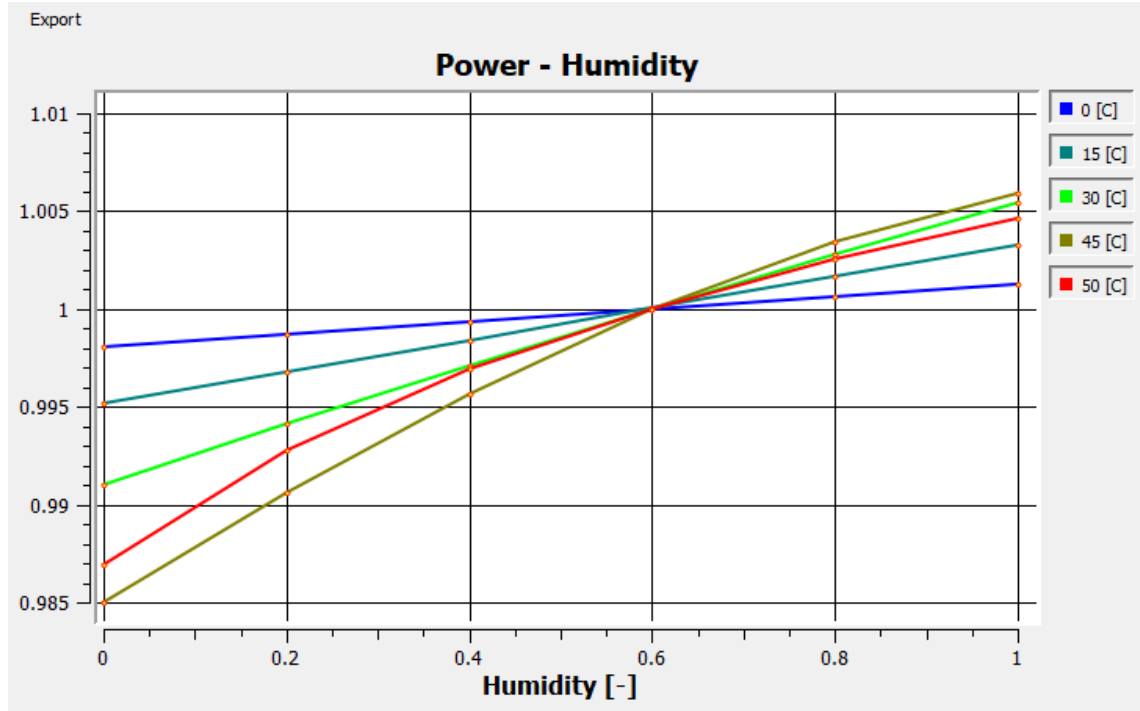


Fig. 6: GT power output correction curve for inlet relative humidity

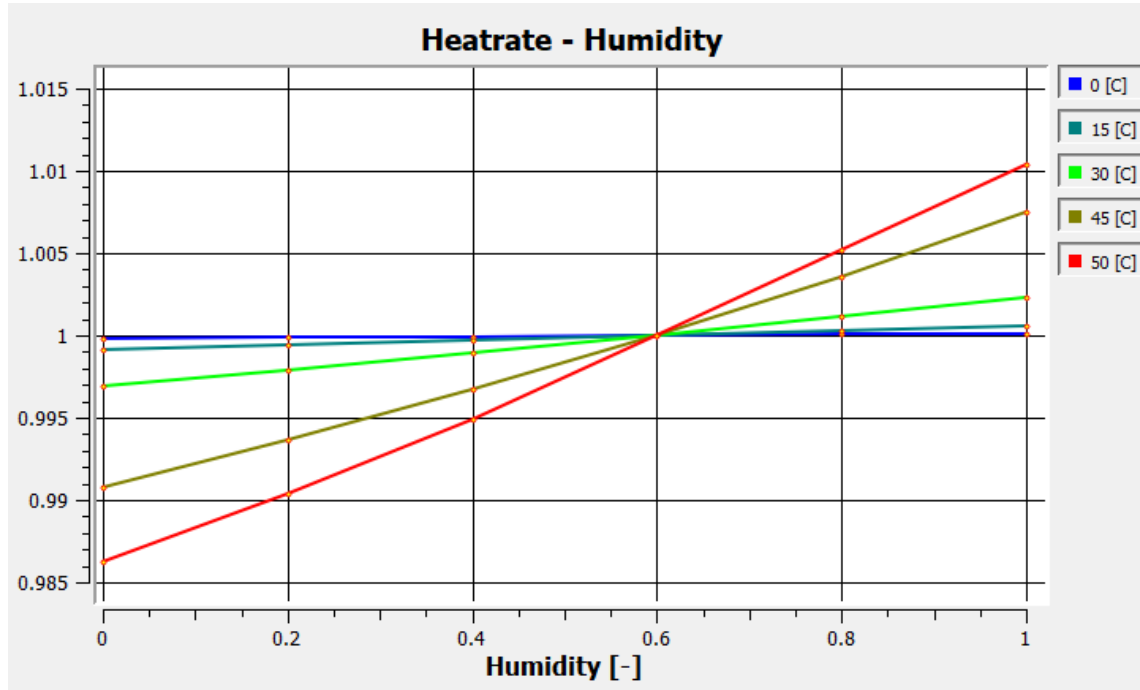


Fig. 7: GT heat rate correction curve for inlet relative humidity

Moreover, the increase of inlet RH decreases the GT exhaust mass because the reduction of the intake air mass is higher than the additional fuel added by the GT controller (Kurz & Brun, 2016). This behavior is represented in Figure 8.

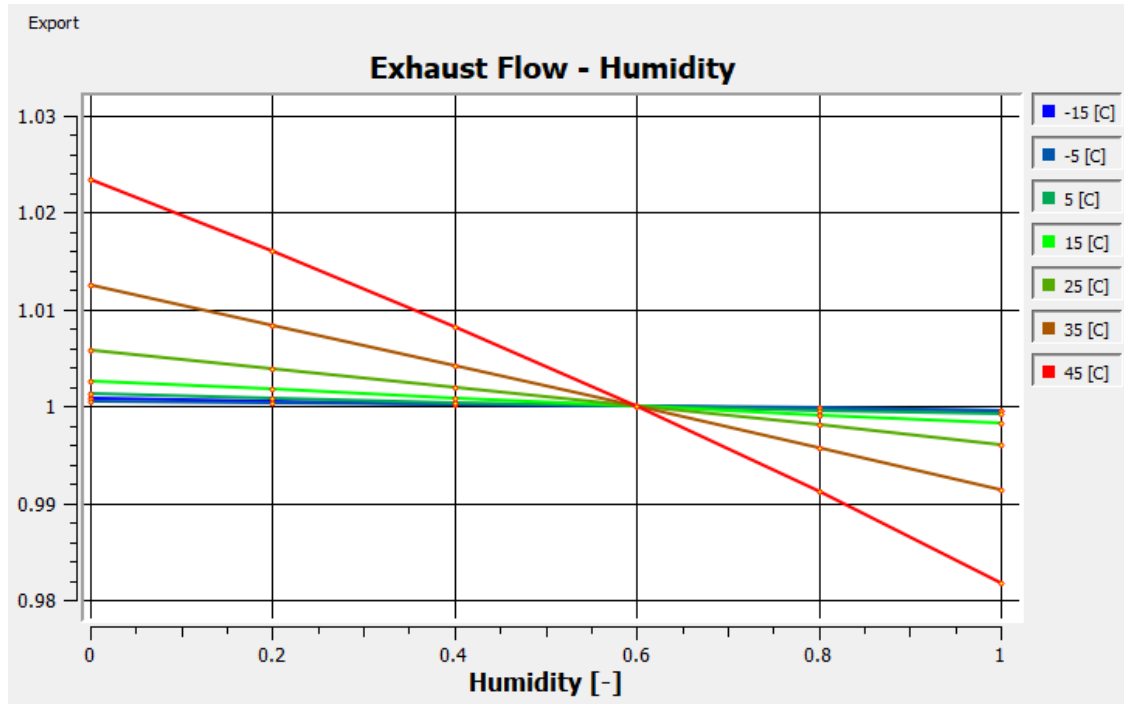


Fig. 8: GT exhaust mass flow correction curve for inlet relative humidity

As discussed earlier, the turbine pressure ratio is inversely proportional to the exhaust temperature after the turbine. The reduction of the air density during the high relative humidity reduces the pressure ratio of the GT. Therefore, the GT exhaust temperature will increase, as seen in Figure 9.

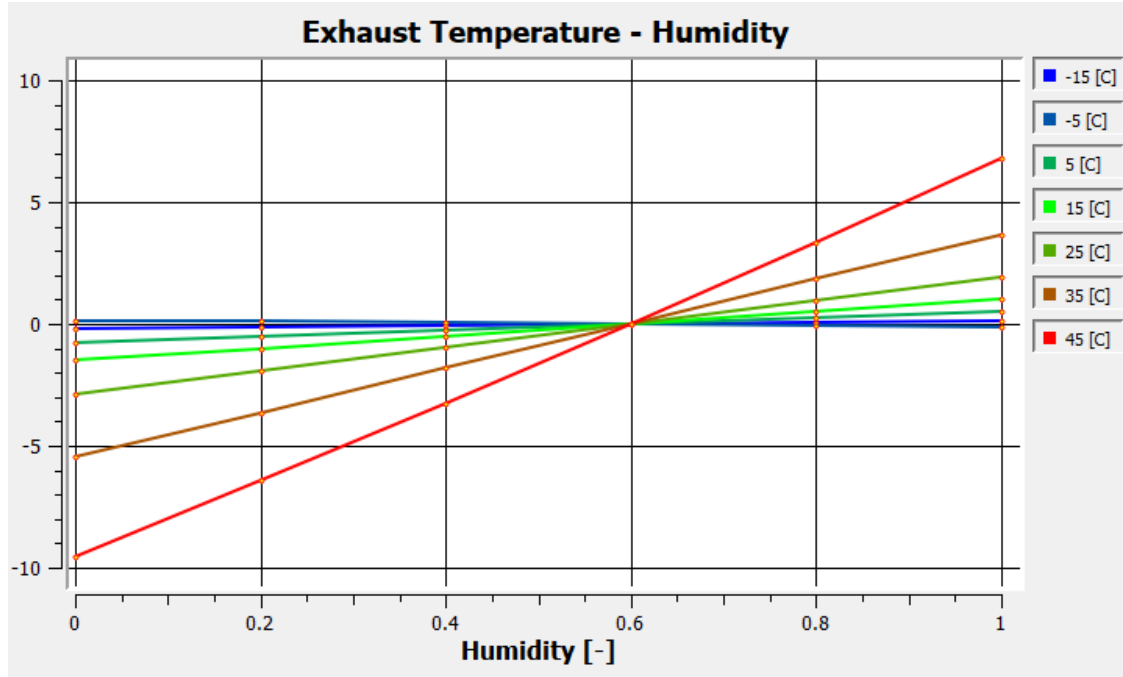


Fig. 9: GT exhaust temperature correction curve for inlet relative humidity

3.2.1.3 Inlet pressure drop correction curves

The intake of the GT involves a filtration system to provide clean air to the GT. The air filtration system causes more restrictions to the airflow, thus reducing the compressor intake pressure. During the machine's long operation, dust accumulates on the air filters, which increases the inlet pressure losses.

Figures 10 and 11 show the inlet pressure drop curves for the output and heat rate, respectively. The increase of inlet pressure drop decreases the GT power output and increases the GT heat rate due to the lower inlet air mass, which decreases the pressure ratio of the GT (Scott et al., 1995).

Moreover, The GT manufacturers provide the machine performance ratings considering a new and clean air filtration system with a fixed-designed inlet pressure drop, which is 9 mbar as per the manufacturer design data for the selected GT.

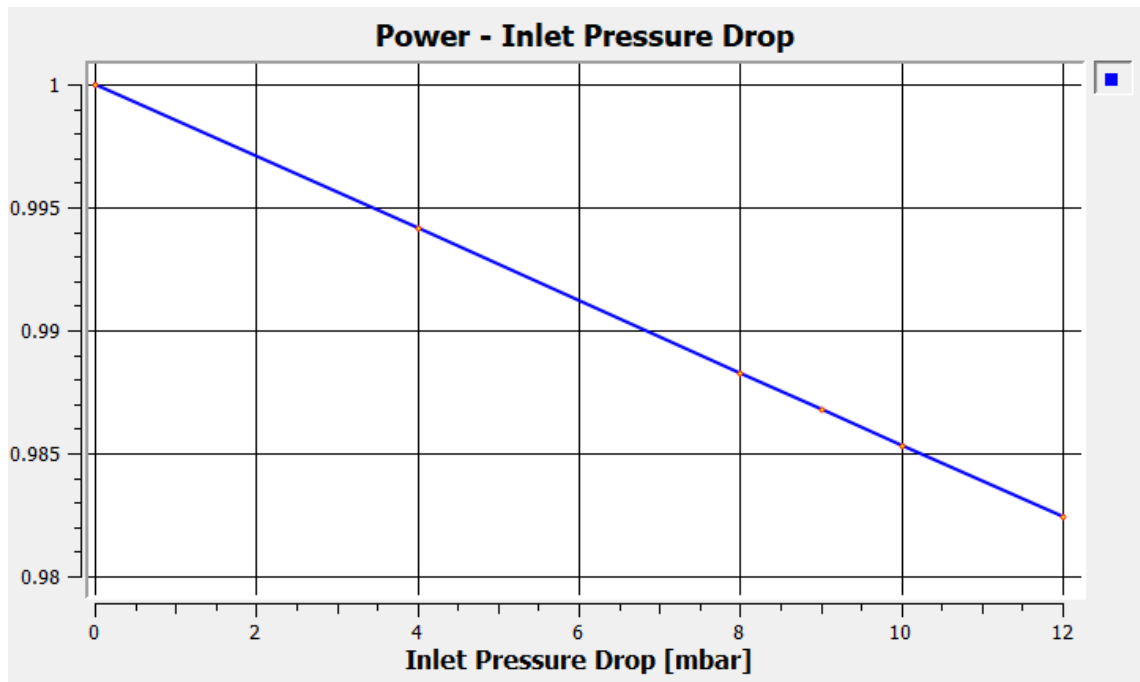


Fig. 10: GT power output correction curve for inlet pressure drop

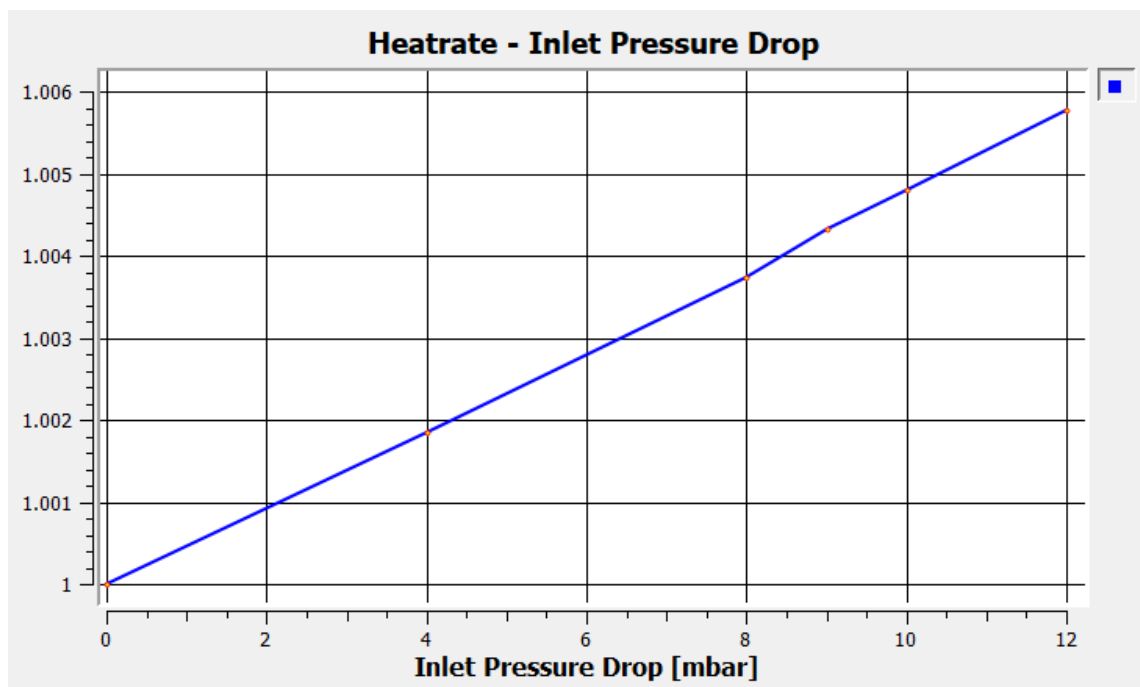


Fig. 11: GT heat rate correction curve for inlet pressure drop

As discussed earlier, the temperature after the turbine is inversely proportional to the GT pressure ratio. Therefore, the increase of inlet pressure drop increases the turbine exhaust temperature, as shown in

Figure 12. In addition, higher intake pressure losses reduce GT exhaust mass flow due to the lesser air intake to the GT, as shown in Figure 13.

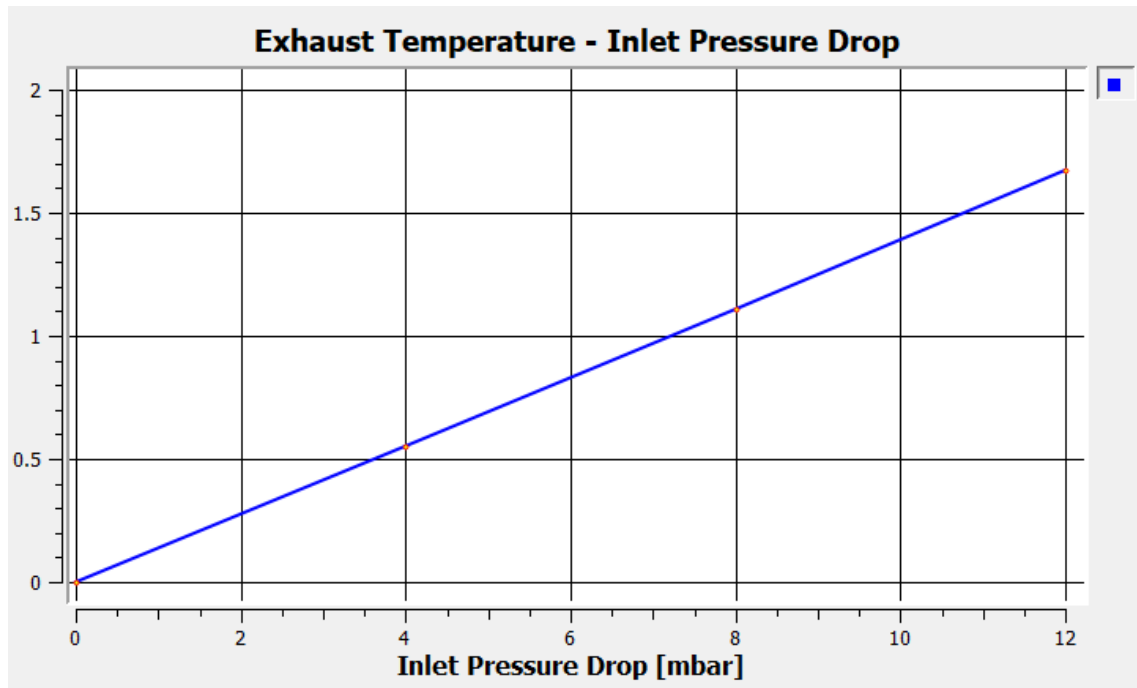


Fig. 12: GT exhaust temperature correction curve for inlet pressure drop

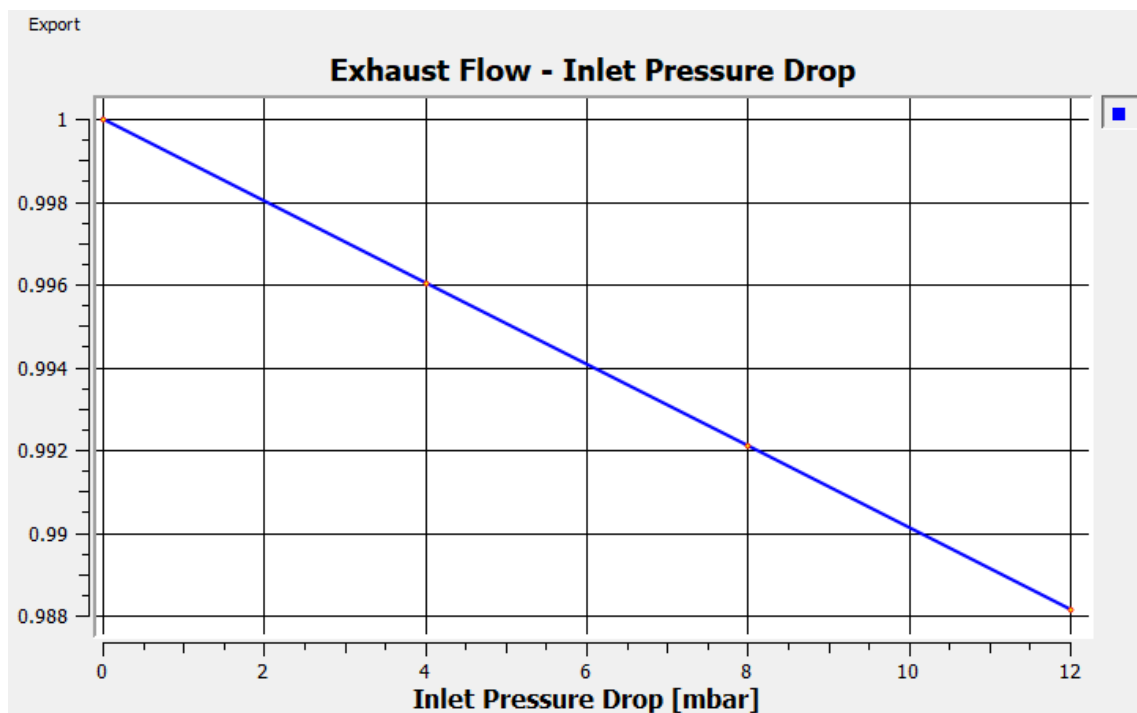


Fig. 13: GT exhaust mass flow correction curve for inlet pressure drop

3.2.1.4 Exhaust pressure drop correction curves

The higher the expansion done by the turbine, the more the generated work. The pressure after the turbine depends on the restrictions in the exhaust duct. The exhaust silencers or HRSG in CC configuration cause more exhaust flow restrictions, especially after long operational periods, which limits the expansion. Therefore, higher exhaust pressure decreases the turbine expansion, which will drop the power output and cycle efficiency (Kurz & Brun, 2016).

The GT exhaust pressure drop is the difference between the surrounding air pressure and the pressure after the turbine. Figures 14 and 15 show the GT power output and heat rate curves for exhaust pressure drop, respectively. It is worth mentioning that the design exhaust pressure drop for the selected GT is 29 mbar, considering a CC operation and clean HRSG tubes.

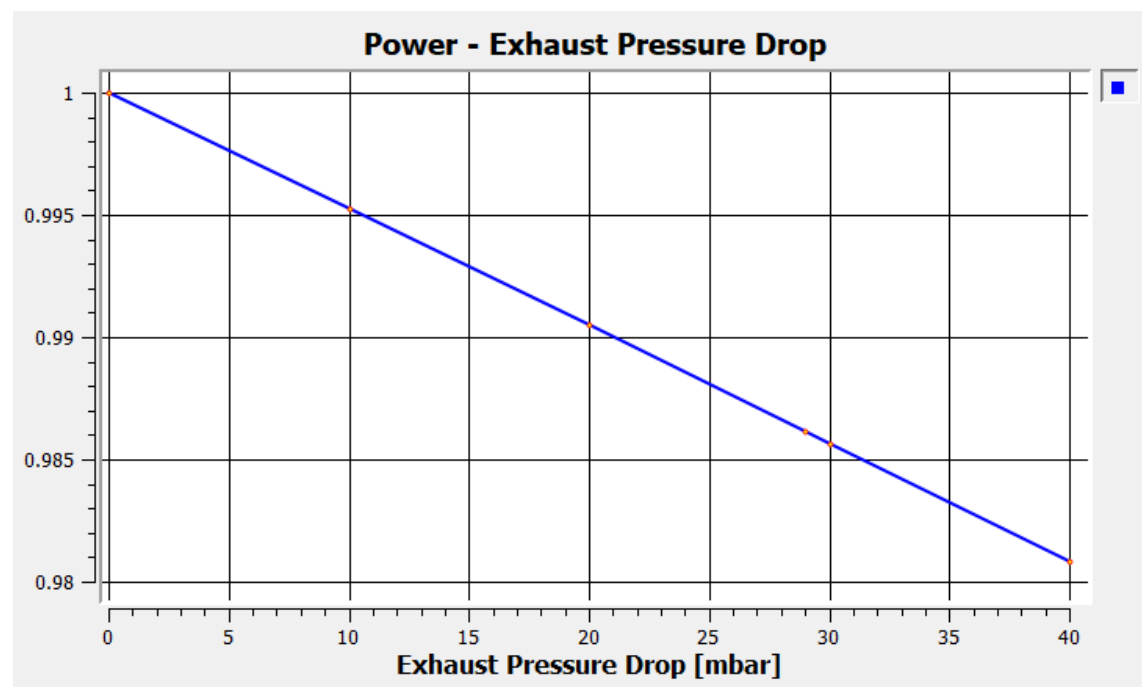


Fig. 14: GT power output correction curve for exhaust pressure drop

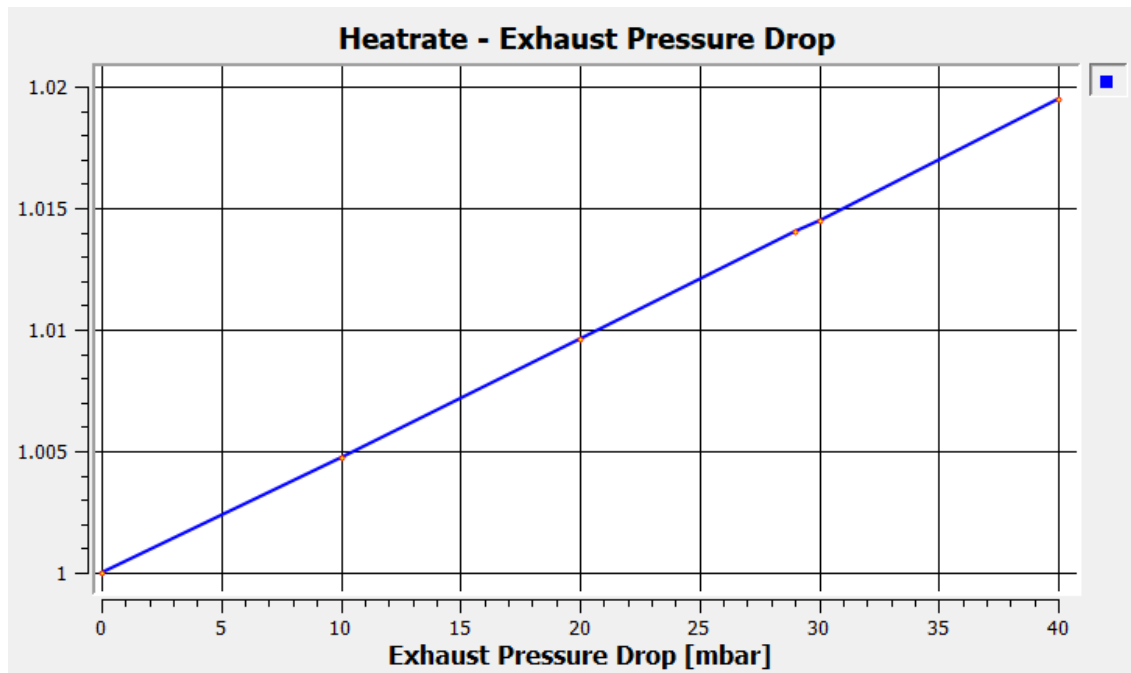


Fig. 15: GT heat rate correction curve for exhaust pressure drop

As the exhaust pressure drop increases, less expansion will be done by the turbine, which increases the exhaust temperature. Figure 16 shows the GT exhaust temperature correction curve for exhaust pressure drop. In addition, the increase in exhaust losses does not affect the exhaust mass flow rate (Scott et al., 1995). Figure 17 shows the GT exhaust mass flow correction curve for exhaust pressure drop. The figure shows that for an exhaust loss of 40 mbar, the exhaust mass flow rate will decrease by around 0.000074%, which is very minor.

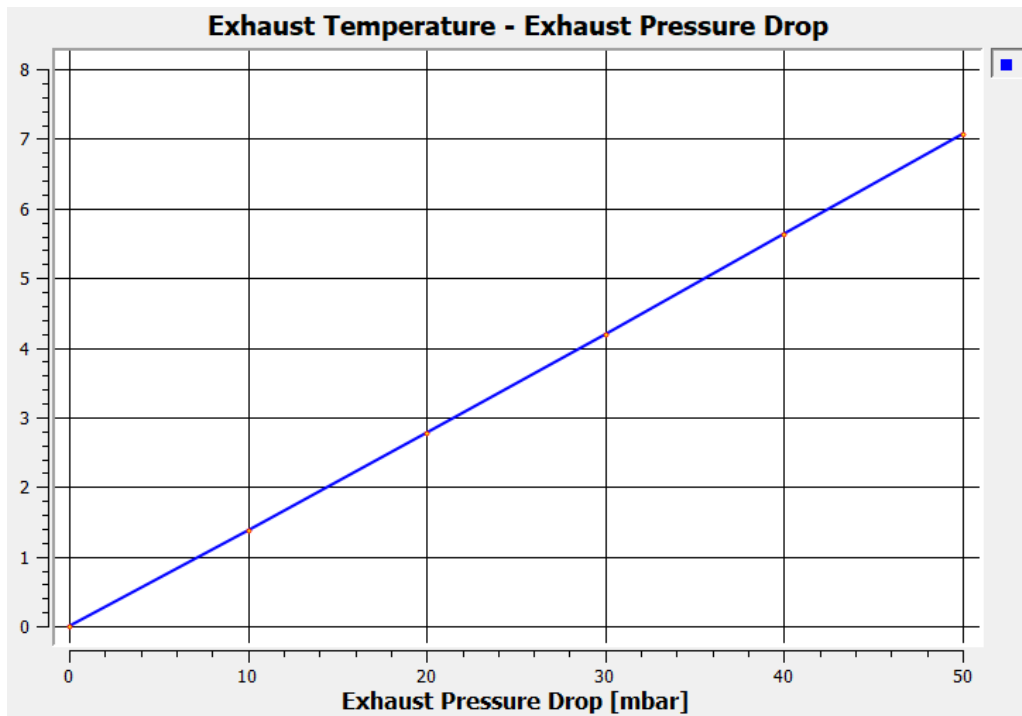


Fig. 16: GT exhaust temperature correction curve for exhaust pressure drop

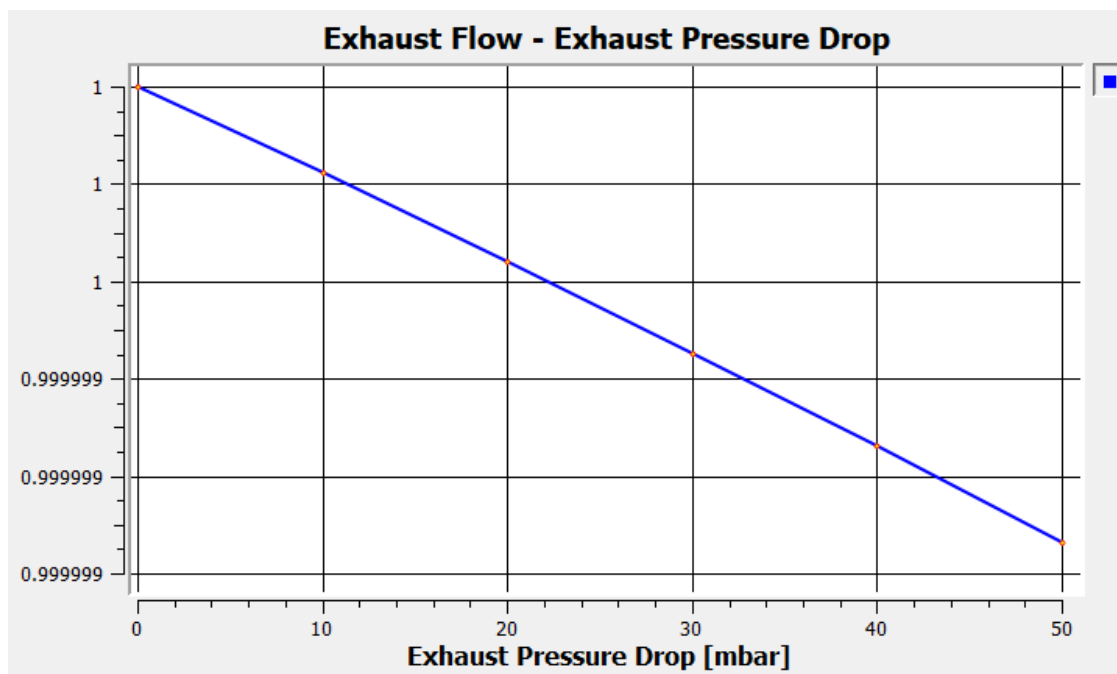


Fig. 17: GT exhaust mass flow correction curve for exhaust pressure drop

3.2.2 Gas turbine model equations

To simulate the yearly machine performance in Bahrain, ambient temperature and relative humidity will vary as per the year 2022 Bahrain's climate data. The input and output differential pressures will be constants per the design values. However, the use of air intake cooling increases the inlet pressure losses. Therefore, the inlet pressure loss will differ per the used cooling system.

Moreover, the ambient pressure, lower gas heating value, frequency, injection flow, and relative power will equal the reference operating condition shown in Table 1. Therefore, their correction curves will not be utilized to build the gas turbine model in Ebsilon.

The overall correction factor for the GT output, heat rate, and exhaust mass flow is given by equations 1-3, respectively (ASME, 2005), (Lee et al., 2017)

$$C.F_P = C.F_{T,P} * C.F_{RH,P} * C.F_{IPD,P} * C.F_{EPD,P} \quad (1)$$

Where:

- $C.F_P$: GT power correction factor
- $C.F_{T,P}$: GT power correction factor for C.I.T
- $C.F_{RH,P}$: GT power correction factor for inlet RH
- $C.F_{IPD,P}$: GT power correction factor for IPD
- $C.F_{EPD,P}$: GT power correction factor for EPD

$$C.F_{HR} = C.F_{T,HR} * C.F_{RH,HR} * C.F_{IPD,HR} * C.F_{EPD,HR} \quad (2)$$

Where:

- $C.F_{HR}$: GT heat rate correction factor
- $C.F_{T,HR}$: GT heat rate correction factor for C.I.T

- $C.F_{RH,HR}$: GT heat rate correction factor for inlet RH
- $C.F_{IPD,HR}$: GT heat rate correction factor for IPD
- $C.F_{EPD,HR}$: GT heat rate correction factor for EPD

$$C.F_{\dot{m}g} = C.F_{T,\dot{m}g} * C.F_{RH,\dot{m}g} * C.F_{IPD,\dot{m}g} * C.F_{EPD,\dot{m}g} \quad (3)$$

Where:

- $C.F_{\dot{m}g}$: Exhaust mass flow correction factor
- $C.F_{T,\dot{m}g}$: GT exhaust gas mass flow rate correction factor for C.I.T
- $C.F_{RH,\dot{m}g}$: GT exhaust gas mass flow rate correction factor for inlet RH
- $C.F_{IPD,\dot{m}g}$: GT exhaust gas mass flow rate correction factor for IPD
- $C.F_{EPL,\dot{m}g}$: GT exhaust gas mass flow rate correction factor for EPD

After calculating the correction factors, the expected values for the output, heat rate, and exhaust mass flow rate at a given operating condition are given by equations 4-6, respectively.

$$P_{GTE} = C.F_P * P_{GTRe} \quad (4)$$

$$HR_{GTE} = C.F_{HR} * HR_{GTRe} \quad (5)$$

$$\dot{m}g_{GTE} = C.F_{\dot{m}g} * \dot{m}g_{GTRe} \quad (6)$$

The correction for the temperature after the turbine is given by equation 7 (ASME, 2005), (Lee et al., 2017).

$$C.F_{TET} = C.F_{T,TET} + C.F_{RH,TET} + C.F_{IPD,TET} + C.F_{EPD,TET} \quad (7)$$

Where:

- $C.F_{TET}$: Turbine exhaust temperature correction factor
- $C.F_{T,TET}$: Turbine exhaust temperature correction factor for inlet air temperature
- $C.F_{RH,TET}$: Turbine exhaust temperature correction factor for inlet relative humidity
- $C.F_{IPD,TET}$: Turbine temperature correction factor for intake pressure drop
- $C.F_{EPD,TET}$: Turbine exhaust temperate correction factor for exhaust pressure drop

The expected value of the turbine exhaust temperature at a given operating condition is given by equation 8:

$$TET_{GTE} = C.F_{TET} + TET_{GTRe} \quad (8)$$

The expected inlet fuel mass flow rate is been calculated using expected values of GT power output and heat rate as follows:

Equation 9 shows the typical GT heat rate equation:

$$HR = \frac{\dot{m}f * LHV * 3600}{P} \quad (9)$$

Where:

- $\dot{m}f$: Inlet fuel mass flow rate (kg/s)
- LHV : Fuel lower heating value (kJ/kg)
- P : GT power output (kW)

The expected inlet fuel mass flow rate at a given operating condition is given by equation 10:

$$\dot{m}f_{GTE} = \frac{HR_{GTE} * P_{GTE}}{LHV * 3600} \quad (10)$$

The expected air intake mass flow is calculated by equation 11:

$$\dot{m}a_{GTE} = \dot{m}g_{GTE} - \dot{m}f_{GTE} \quad (11)$$

It is worth mentioning that the selected plant is using khuff gas as an inlet fuel for the gas turbines. The HHV and LHV of the used khuff gas are given in Table 3.

Table (3): Khuff Gas Higher and Lower Heating Values

Khuff Gas Heating Values	
HHV (kJ/kg)	38001.59
LHV (kJ/kg)	34263.02

3.3 HRSG Modeling

HRSG is the equipment that connects GT and ST cycles, by using the heat available in the gas after the GT for steam formation. As discussed in the GT modeling section, the operating parameters affect the gas specifications after the turbine, which affects the input heat energy for the HRSG.

High pressure circuit process

As seen in Figure 1, the HP feed water is first supplied to four HP economizers in series (HPECO1, HPECO2, HPECO3, HPECO4). The function of the economizers is only to increase the water temperature near the saturation temperature in the HP evaporator (HPEVA). The function of the evaporator is to change the water phase from liquid to steam. Once the steam forms in the evaporator, it enters three superheaters in series (HPSH1, HPSH2, and HPSH3), which increases the steam temperature to the design outlet temperature before it is admitted to the ST.

Low pressure circuit process

The LP water is first supplied to the LP economizer (LPECO), as shown in Figure 1. After the economizer, a fixed amount of water is transferred to the deaerator, while the remaining is sent to the LP evaporator (LPEVA), where the steam is formed. Finally, the steam enters the superheater (LPSH) before sending it to the LP steam common header.

3.3.1 HRSG modeling in Ebsilon

The HRSG has been modeled in Ebsilon based on the energy balance and heat transfer methods of each individual heat exchanger in the HRSG (Ahmed et al., 2018), (Ali et al., 2017), (Eriksen & Schroeder, 2017).

Unfortunately, the availability of HRSG design data is limited, so the remaining unavailable has been set as per the recommended values by Ebsilon. Equations 12-20 show the basis of modeling the HRSG in Ebsilon. The amount of heat that has been released from the gas to the water or steam in the HRSG section is given by equations 12 and 13, based on the LMTD method:

$$\dot{Q}_{released} = U A LMTD \quad (12)$$

$$LMTD = \frac{(Th,o - Tc,i) - (Th,i - Tc,o)}{\ln \frac{(Th,o - Tc,i)}{(Th,i - Tc,o)}} \quad (13)$$

Where:

- $\dot{Q}_{released}$: Heat transfer rate that has been released (kW)
- U: Heat transfer coefficient (kW/m²K)
- A: Surface area (m²)
- LMTD: Log mean temperature difference (K)
- Th,o : Hot stream outlet temperature (K)
- Th,i: Hot stream inlet temperature (K)

- $T_{c,o}$: Cold stream outlet temperature (K)
- $T_{c,i}$: Cold stream inlet temperature (K)

The amount of heat that has been released from the gas to the water or steam in any heat exchanger in the HRSG can be also calculated based on energy balance principles given by equation 14:

$$\dot{Q}_{released} = \dot{m}_g (h_{g_{in}} - h_{g_{out}}) = \dot{m}_g C_p(T_{g_{in}} - T_{g_{out}}) \quad (14)$$

Where:

- $\dot{m}_g$: GT exhaust gas mass flow rate (kg/s)
- $h_{g_{in}}$: The inlet enthalpy of the gas (kJ/kg)
- $h_{g_{out}}$: The outlet enthalpy of the gas (kJ/kg)
- C_p : Gas specific heat capacity (kJ/kg.K)
- $T_{g_{in}}$: The temperature of the inlet gas (K)
- $T_{g_{out}}$: The temperature of the outlet gas (K)

The absorbed heat by the water or steam in any heat exchanger in the HRSG depends on the heat exchanger heat losses and it can be determined by equation 15:

$$\dot{Q}_{absorbed} = \dot{Q}_{released} * (1 - HL) \quad (15)$$

- $\dot{Q}_{absorbed}$: The heat that the steam or water absorbs (kW)
- $\dot{Q}_{released}$: The heat that the exhaust gas emitted (kW)
- HL : Heat exchanger heat loss

The amount of steam that has been generated in the evaporator is given by equations 16 and 17 as follows (Eriksen & Schroeder, 2017):

$$\dot{m}_s = \frac{\dot{Q}_{absorbed}}{\Delta h_s} \quad (16)$$

$$\Delta h_s = h_{s,out} - h_{w,in} \quad (17)$$

Where:

- $\dot{m}_s$: Produced steam mass flow rate (kg/s)
- Δh_s : The energy required to produce 1 kg/s of steam (kJ/kg)
- $h_{s, out}$: Outlet steam enthalpy (kJ/kg)
- $h_{w, in}$: Inlet water enthalpy (kJ/kg)

Table 4 illustrates the design input data of each section of the HRSG. Moreover, Table 5 shows the recommended values by Ebsilon, which were not available in the design documents.

Table (4): HRSG Design Data

Heat exchanger	Heat Transfer Surface Area (m ²)	Gas side Heat transfer coefficient (W/m ² K)	Water/steam side pressure drop (bar)	Gas side pressure drop (mbar)
High pressure circuit				
ECO 1	7971	43.41	4	1.9
ECO 2	3985	39.65	0.8	0.7
ECO 3	15941	46.36	2.4	2.9
ECO 4	7971	45.92	4	1.9
EVA	30017	52.04	25.2	8.5
SH 1	4872	37.36	0.8	1.6
SH 2	4872	36.33	0.6	1.7
SH 3	4872	36.33	1.1	1.7
Low pressure circuit				
ECO	4288	43.38	0.8	0.7
EVA	15941	43.10	11.3	3.5
SH	2610	26.82	0.2	0.7

Table (5): Default HRSG Data as Per Ebsilon

Heat exchanger	Water side Heat transfer coefficient (W/m ² K)	Water side mass flow rate exponent	Gas side mass flow rate exponent	Heat loss (%)
High Pressure Circuit				
ECO 1	500	0.784	0.61	0.5
ECO 2	500			
ECO 3	500			
ECO 4	500			
EVA	10000			
SH 1	800			
SH 2	800			
SH 3	800			
Low Pressure Circuit				
ECO	500	0.784	0.61	0.5
EVA	10000			
SH	800			

During the off-design, the exhaust gas parameters change. These changes affect HRSG heat transfer coefficients. Therefore, the generated steam parameters will change too. (Ghiasi et al., 2021) and (Delgado Ruiz et al., 2018) investigated the impact of fluid mass flow rates on the overall amount of heat transferred and the heat transfer coefficients. The studies agreed that by increasing the mass flow rates of either side of a heat exchanger, the heat transferred and the heat transfer coefficients increase, too.

Ebsilon software uses the flow rate exponent to estimate the off-design behavior of the HRSG (EBSILON®Professional, n.d.). The effect of the mass flow rate and heat transfer coefficient is governed by the mass flow

rate exponent. Therefore, whenever the exhaust gas parameters change, the software calculates the off-design heat transfer coefficients of each section in the HRSG.

The off-design heat transfer coefficients of each section in the HRSG are calculated in Ebsilon by the following equations:

Cold side (Water):

$$U_{od} = U_d * \left(\frac{m_{od}}{m_d} \right) * EX_{cs} \quad (18)$$

Where:

- U_{od} : Cold side heat transfer coefficient during off design condition (kW/m²K)
- U_d : Cold side heat transfer coefficient during design condition (kW/m²K)
- m_{od} : Cold side mass flow during off design condition (kg/s)
- m_d : Cold side mass flow rate during design condition (kg/s)
- EX_{cs} : Mass flow rate exponent of the cold side

Hot side (Exhaust Gas):

$$U_{od} = U_d * \left(\frac{m_{od}}{m_d} \right) * EX_{hs} * (1 - 0.0005 * (Tm_d - Tm_{od})) \quad (19)$$

$$Tm = \frac{T_{hs,in} + T_{hs,out}}{2} \quad (20)$$

Where:

- U_{od} : Hot side heat transfer coefficient during off design condition (kW/m²K)
- U_d : Hot side heat transfer coefficient during design condition (kW/m²K)
- m_{od} : Hot side mass flow rate during off design condition (kg/s)

- m_d : Hot side mass flow rate during design condition (kg/s)
- EX_{hs} : Mass flow rate exponent of the hot side
- Tm : Mean temperature of outlet and inlet (K)
- Tm_d : The design mean temperature of outlet and inlet (K)
- Tm_{od} : The off design mean temperature of outlet and inlet (K)

3.4 Steam Turbine Modeling

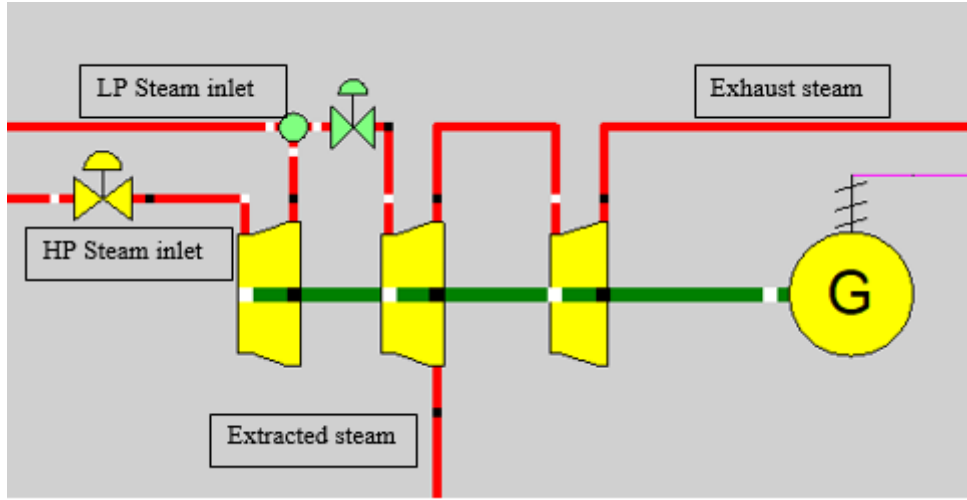


Fig. 18: Steam turbine model in Ebsilon

The selected ST for this research has one HP steam inlet and one LP steam inlet, as shown in Figure 18. Moreover, it has also one steam extraction to the feed water preheater, as discussed earlier.

As per the Ebsilon user manual, if there are any admissions or extractions of steam during the expansion process, the process should be modeled using multiple ST components (EBSILON®Professional, n.d.).

Therefore, the ST has been modeled in Ebsilon using three ST components. Each single component is considered to be a group of expansion stages.

The generated power of the ST component is given by:

$$P_{st} = (\dot{m}_s * (hs_{inlet} - hs_{outlet,isen}) * \eta_{isen}) * \eta_{mech} * \eta_{Gen} \quad (21)$$

Where:

- P_{st} : Steam turbine power output (kW)
- $\dot{m}_s$: Mass flow rate of the inlet steam (kg/s)
- $h_{s\ inlet}$: The enthalpy of the inlet steam (kJ/kg)
- $h_{s\ outlet, isen}$: Isentropic process outlet steam enthalpy (kJ/kg)
- η_{isen} : ST isentropic efficiency
- η_{mech} : ST mechanical efficiency
- η_{Gen} : Generator efficiency

The input design values of the selected steam turbine are shown below in table 6:

Table (6): Selected steam turbine design values

Parameter	Value
Inlet HP steam pressure (bar)	64.2
Inlet LP steam pressure (bar)	4.5
Extracted steam pressure (bar)	0.813
Exhaust steam pressure (bar)	0.3
Isentropic efficiency (%)	91.0
Mechanical efficiency (%)	99.8
Generator efficiency (%)	98.6

Since the amount of steam that flowed through the ST is directly proportional to the isentropic efficiency (Fraidenraich et al., 2013), recommended values by Ebsilon have been used to increase the accuracy of the off-design simulations of the ST model. Table 7 shows the used isentropic efficiency correction values.

Table (7): Steam Turbine Isentropic Efficiency Correction Data as Per Ebsilon

$\frac{mS_{od}}{mS_d}$	$\frac{\eta_{isen\ od}}{\eta_{isen\ d}}$
0	0.98
0.5	0.991
0.6	0.993
0.7	0.995
0.8	0.9975
0.9	0.999
0.95	0.9998
1	1
1.05	0.9995
1.1	0.998

3.5 Deaerator Modeling

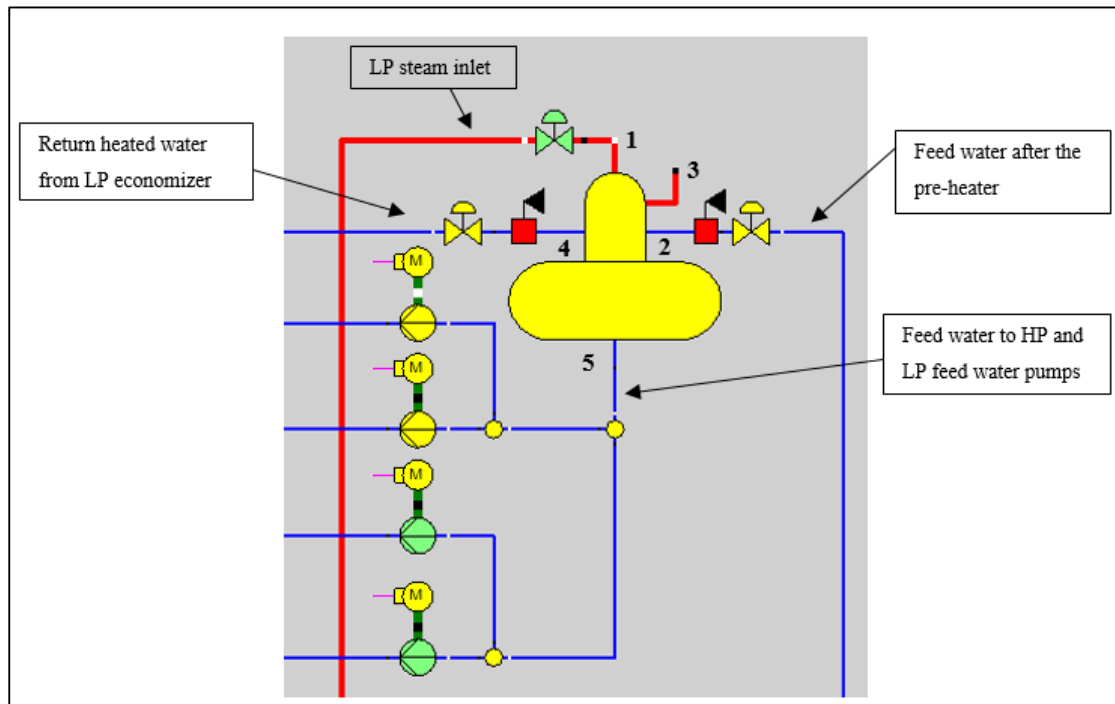


Fig. 19: Deaerator model in Ebsilon

The deaerator is used in the ST cycle to separate the non-condensable gasses from the feed water before it is supplied to the HRSGs. The

separation mechanism in the deaerator uses an LP super-heated steam to heat the feed water to remove the non-condensable gases through a vent. The deaerator has been designed in a way that the supplied steam will increase the feed water until it has been fully condensed. The deaerators are an open-feed water heater system where the used steam and the feed water steam are mixed. Figure 19 shows the Deaerator model in Ebsilon, which is modeled based on energy and mass balance. Table 8 contains the design values for the deaerator.

Table (8): Deaerator Design Parameters

Deaerator Pressure (bar)	Deaerator Temperature (°C)	The mass flow rate of return water to LP economizers (kg/s)	Vented steam mass flow rate (kg/s)
3.13	135	60	0

The mass balance for the deaerator is given by equation 22:

$$\dot{m}_1 + \dot{m}_2 + \dot{m}_4 = \dot{m}_5 + \dot{m}_3 \quad (22)$$

Where:

- $\dot{m}_1$: Mass flow rate of the inlet LP steam (kg/s)
- $\dot{m}_2$: Mass flow rate of the feed water (kg/s)
- $\dot{m}_3$: Mass flow rate of the vented steam (kg/s)
- $\dot{m}_4$: Mass flow rate of the return water (kg/s)
- $\dot{m}_5$: Mass flow rate of the total water flow to the HRSG (kg/s)

The steam requirement by the deaerator is given by the energy balance as follow:

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 + \dot{m}_4 h_4 = \dot{m}_5 h_5 + \dot{m}_3 h_3 \quad (23)$$

$$\dot{m}_1 = \frac{\dot{m}_5 h_5 + \dot{m}_3 h_3 - \dot{m}_2 h_2 - \dot{m}_4 h_4}{h_1} \quad (24)$$

3.6 Feed Water Pre-Heater Modeling

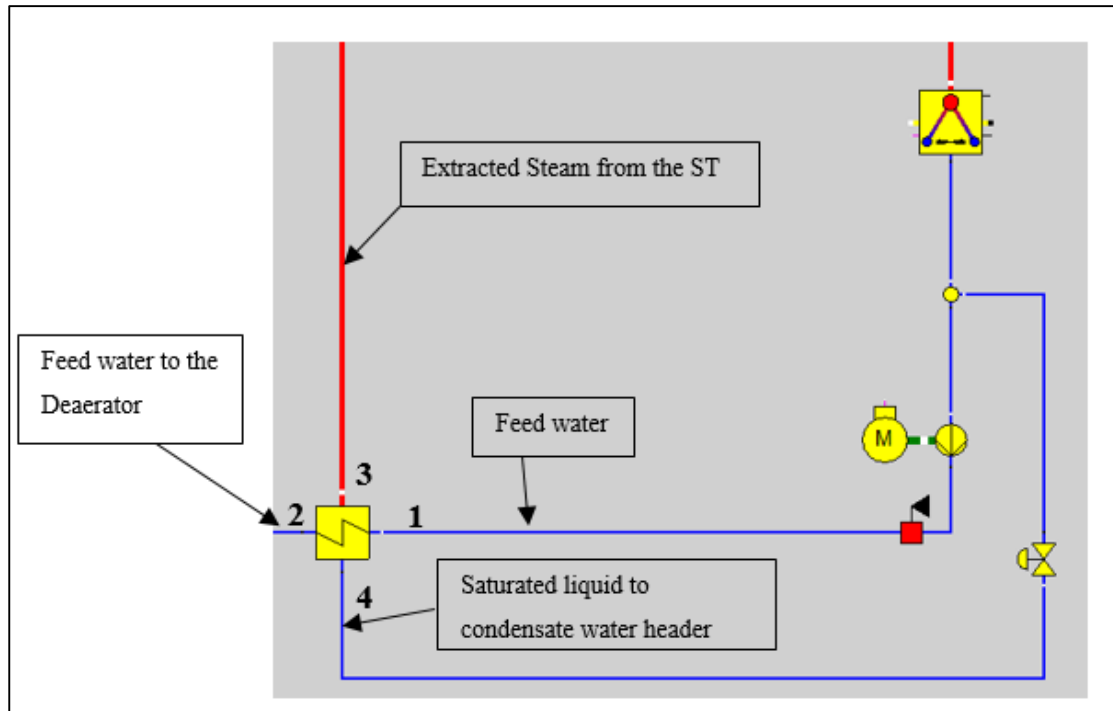


Fig. 20: Feed water pre-heater model in Ebsilon

The feed water preheater is a heat exchanger used to increase the temperature of the water before it is supplied to the deaerator to increase the cycle efficiency (Mahendra et al., 2021). The used feed water preheater in the selected plant is a closed feedwater heater, with no mixing between the cold and hot streams. The preheater uses extracted LP steam from the ST as a hot stream, where it is condensed after the water is heated.

The feed water preheater has been modeled in Ebsilon using the upper terminal temperature difference (UTTD) method using the manufacturer's data. The UTTD is the difference between the hot supplied extracted steam and the water temperature flow to the deaerator. Moreover, the amount of the extracted steam mass flow depends on the required UTTD and the

water inlet temperature at port 1, which depends on the steam turbine back pressure. The calculation in Ebsilon for the feed water preheater is based on mass and energy balance equations as follows:

Mass balance:

$$\dot{m}_1 + \dot{m}_3 = \dot{m}_2 + \dot{m}_4 \quad (25)$$

Where:

- $\dot{m}_1$: The mass flow of the inlet water (kg/s)
- $\dot{m}_2$: The mass flow of the outlet water (kg/s)
- $\dot{m}_3$: The mass flow of the extracted steam (kg/s)
- $\dot{m}_4$: The mass flow rate of the saturated liquid to condensate water header (kg/s)

Energy balance:

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (26)$$

Required steam to achieve the design UTTD:

$$\dot{m}_3 = \frac{\dot{m}_2 h_2 + \dot{m}_4 h_4 - \dot{m}_1 h_1}{h_3} \quad (27)$$

The upper terminal temperature difference:

$$UTTD = T_3 - T_2 \quad (28)$$

Table 9 below includes the design values of the feed water pre heater.

Table (9): Design values of the feed water pre heater

Design Parameter	Value
UTTD	3.32 K
Extracted steam pressure	0.813 bar
Extracted steam Temperature	93.92 °C

3.7 Air Cooled Condenser Modeling

One of the most critical equipment in the ST plants is the condenser. The condenser in the ST plant is used to transform the steam after the turbine from the steam phase to the water phase by rejecting its latent heat using a cold stream of water or ambient air. For the selected plant, the condenser is an A-shape air-cooled condenser that uses ambient air to condense the exhaust steam from the ST. Figure 21 shows a cross-section overview of the condenser used in the selected CCGT plant.

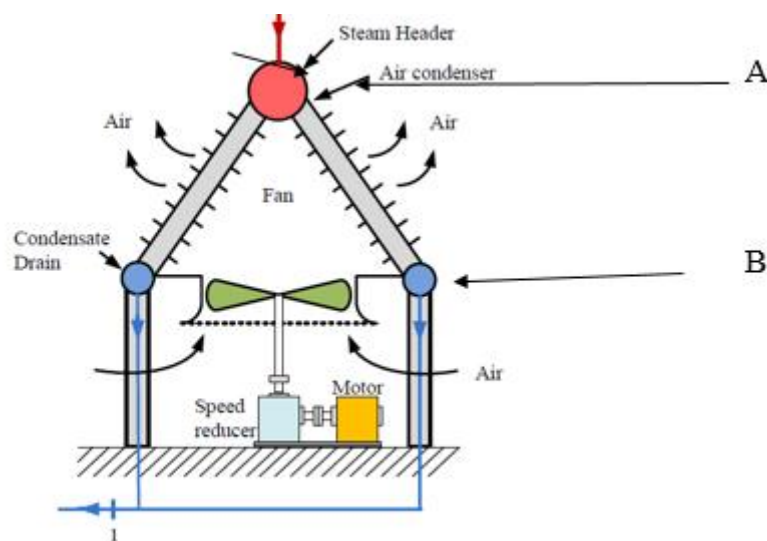


Fig. 21: Cross section view of an A-Shape air cooled condenser:
Kroger, D. G. (2004). *Air-cooled heat exchangers and cooling towers: thermal-flow performance evaluation and design* (Doctoral dissertation, Stellenbosch: Stellenbosch University).

In the A-shape air-cooled condensers, the exhaust steam is first distributed on main steam headers, which will be distributed in finned tubes that make the A roof shape, as shown in Figure 21. The ambient air enters by axial flow fans to flow through the finned tubes that contain the exhaust steam. As the exhaust steam travels from point A to point B, it condenses.

The input air temperature impacts the air-cooled condensers' performance because their heat removal capacity decreases at high inlet air temperatures. The result is a rise in condenser pressure, which decreases the efficiency and power production of the ST plant (Pattanayak et al., 2019).

The air-cooled condenser has been modeled in Ebsilon based on the design condenser pressure curves collected from the manufacturer's data, as shown in Figure 22.

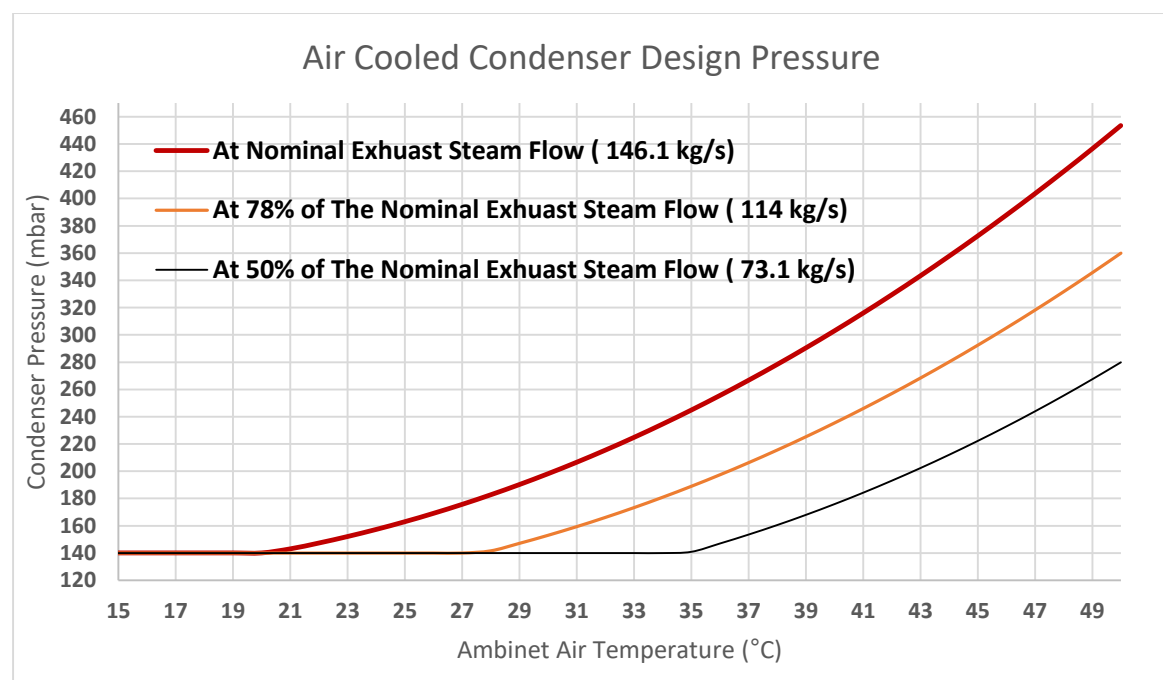


Fig. 22: Air cool condenser design pressure curves

Based on the curves in Figure 22, the condenser pressure is a function of both the amount of steam leaving the ST and the inlet air temperature. As per the ACC design that has been selected, the condenser pressure set point is 140 mbar, maintained by the condenser controller. The controller maintains the pressure by controlling the number of axial flow fans in service as per the condenser pressure input signal. During low ambient temperatures, the condenser has a design capacity to condense the steam with a pressure lower than the design value. However, the controller

reduces the number of fans in service to avoid operating at a pressure below the set point. A similar methodology applies during high ambient temperatures, where the controller will increase the number of fans in service to maintain the pressure at its set point. Unfortunately, the condenser's ability to keep the pressure at the set point is limited during high ambient temperatures due to the limited number of fans. This effect can be seen clearly by referring to Figure 22. For instance, when the steam turbine operates at the nominal exhaust steam flow, the condenser can keep the pressure at the design value of 140 mbar until the air reaches 21 °C.

3.8 Water Pumps Modeling

The pumps in the selected CCGT plant have been used in two locations per the plant layout in Figure 1. Firstly, after the condenser, the condensate extraction pump (CEP) is used to pump the water to the deaerator. Secondly, after the deaerator, HP and LP feed water pumps feed the HRSG with the required feed water. Table 10 includes the design specifications used to model the pumps in Ebsilon.

Table (10): Design Specifications of the Used Pumps in the Selected Plant

Pump	Inlet Pressure (Bar)	Discharge Pressure (Bar)	Isentropic Efficiency (%)	Motor Electrical Efficiency (%)
HP FWP	3.13	105.00	85	95
LP FWP	3.13	17.20	80	95
CEP	0.30	14.00	80	97

3.9 Combined Cycle Performance

In the earlier sections, it has been discussed the modeling methodology of the selected combined cycle plant. This section will discuss the formulation used to calculate the total plant's gross power output and efficiency.

The total plant generation is calculated by using Equation 29:

$$P_{CC} = P_{GT1} + P_{GT2} + P_{ST} \quad (29)$$

Since both gas turbines are identical, the total generation is given by:

$$P_{CC} = 2P_{GT} + P_{ST} \quad (30)$$

The combined cycle thermal efficiency is given by:

$$\eta_{th,LHV} = \frac{P_{CC}}{2 \cdot \dot{m}f \cdot LHV} \quad (31)$$

Where:

- P_{CC} : Total combined cycle generated power (kW)
- $\dot{m}f$: The fuel inlet mass flow rate per GT (kg/s)
- LHV : Lower heating value of the used gas (kJ/kg)

It is worth mentioning that the LHV will be used to calculate the efficiency of the combined cycle plant, while HHV will be used to calculate the total gas cost for the economical part of this research.

3.10 Gas Turbine Inlet Air Cooling Systems

The performance of the GT is sensitive to the operating conditions, especially the air intake temperature, as discussed earlier in the GT modeling part. Bahrain is located in the Middle East, where the ambient temperatures can reach up to 48 °C. Thus, the GT capacity and efficiency drop dramatically. To reduce the effect of high ambient temperatures, an inlet air cooling system should be installed.

The cooling methods used for gas turbine air intake are categorized into two main categories:

1. The evaporative cooling
2. The refrigeration cooling

The evaporative cooling systems, while the refrigeration cooling is not. In this research, four different cooling systems were used. From the evaporative cooling category, the fogging and evaporative cooler systems were used, while from the refrigeration cooling category, the mechanical and absorption chillers were used. The following part will discuss the methodology behind each cooling system.

3.10.1 Evaporative cooling

Evaporative cooling is a process based on utilizing the sensible heat of air to supply the required latent heat to evaporate the water added to the process. It is an adiabatic saturation process, where the heat exchange between the air stream and its surroundings is neglected. In addition, it can be assumed that the air enthalpy during the evaporative cooling is constant. Therefore, the process is based on a constant enthalpy line in the psychrometric chart (Çengel & Boles, 2015). Figure 23 shows the evaporative cooling process on the psychrometric chart. The relative humidity and the specific humidity are increasing during the evaporative cooling process, which is due to water evaporation.

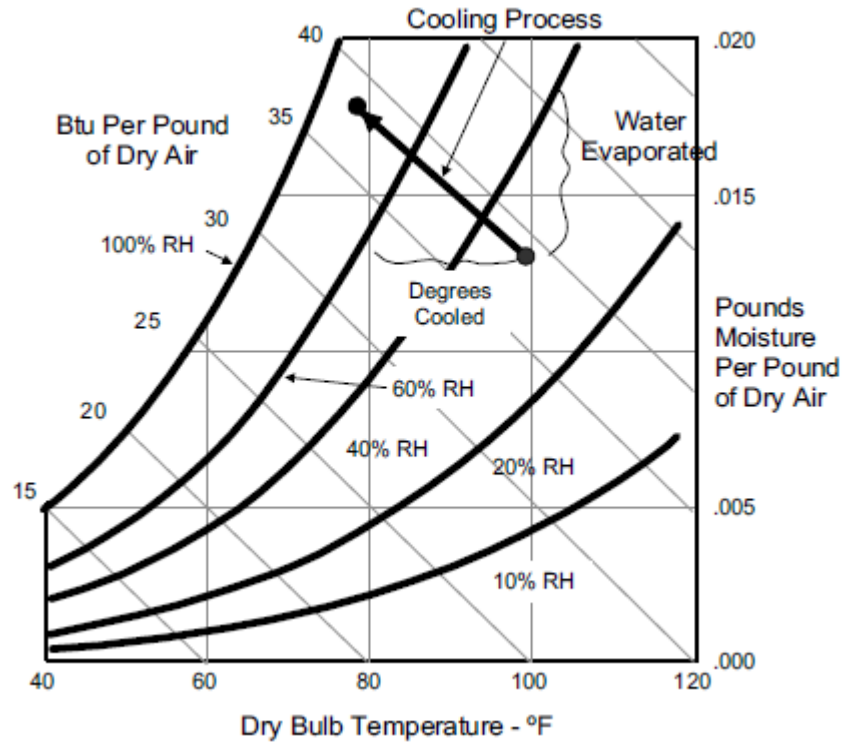


Fig. 23: Evaporative cooling process on psychrometric chart: Jones, C., & Jacobs III, J. A. (2000). Economic and technical considerations for combined-cycle performance-enhancement options. *GE Power Systems, GER-4200*.

The minimum air temperature that could be reached by an ideal evaporative cooling process is the wet bulb temperature. The wet bulb temperature is the air temperature at a relative humidity of 100%, where the air cannot absorb more water vapor. Moreover, the amount of air temperature reduction by the evaporative cooling is limited by its initial relative humidity. The lesser the relative humidity, the more air temperature reduction occurs.

The effectiveness of a cooling system based on an evaporative cooling process is measured by a ratio between the actual temperature drop and the ideal temperature drop called effectiveness. The effectiveness is given by equation 32.

$$Effectiveness = \frac{T_{in,DB} - T_{out}}{T_{in,DB} - T_{WB}} \quad (32)$$

Where:

- $T_{in,DB}$: The inlet air dry bulb temperature (°C)
- T_{out} : The outlet air temperature (°C)
- T_{WB} : The air wet bulb temperature (°C)

The evaporated water mass flow rate during the evaporative cooling process is calculated by applying the mass water balance equation as follows:

$$\dot{m}_{da1}\omega_1 + \dot{m}_{eva.water} = \dot{m}_{da2}\omega_2 \quad (33)$$

Where:

- $\dot{m}_{da1}$: The dry air mass flow rate before cooling (kg/s)
- $\dot{m}_{da2}$: The dry air mass flow rate after cooling (kg/s)
- ω_1 : The air specific humidity before cooling (kg of water/kg of dry air)
- ω_2 : The specific humidity after the cooling (kg of water/kg of dry air)
- $\dot{m}_{eva.water}$: The mass flow rate of the evaporative water during the cooling process (kg/s)

Since no air has been entered or left the system:

$$\dot{m}_{da1} = \dot{m}_{da2} = \dot{m}_{da} \quad (34)$$

Therefore, the amount of water that has been evaporated is given by equation 35:

$$\dot{m}_{eva.water} = \dot{m}_{da}(\omega_2 - \omega_1) \quad (35)$$

Fogging system

The fogging system is one of the most famous systems that has been used to minimize the effect of the C.I.T. The fogging system is based on evaporative cooling, where water is sprayed after the air intake filters through nozzles to form a real fog in the air inlet duct. Thus, the hot air will be cooled by evaporating the fog molecules. Figure 24 shows a typical gas turbine air intake fogging system.

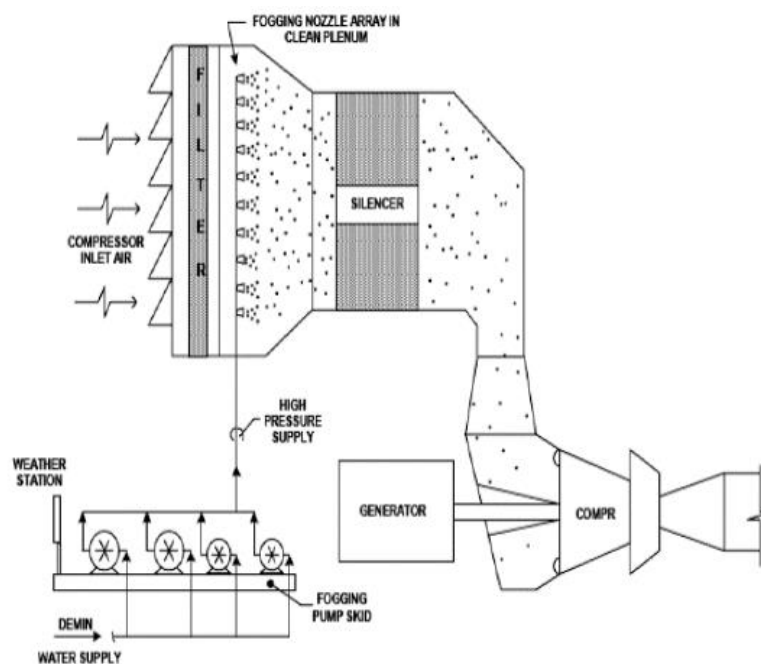


Fig. 24: Air Intake Fogging System: Sakhaei, S. A., & Safari, M. (2014). Study and comparison of inlet air cooling technique of gas turbines and their effects on increase of the efficiency and outlet power. *International Journal of materials, mechanics and manufacturing*, 2(4), 329-334.

The advantage of the fogging system is that the formed fog creates a large evaporative surface, which can cool the inlet air close to its wet bulb temperature. Therefore, the fogging system effectiveness could be assumed to be 100% (Ghosh et al., 2015), (Renzi et al., 2014). In addition, the fog nozzles should be capable of spraying the water molecules with a diameter

of 40 micrometers or less to ensure a suitable fog formation and, therefore, maximum possible effectiveness (Jones & Jacobs, 2000).

The demineralized water is used for the fogging system, where almost all its minerals and impurities have been removed. The reasons are to protect the air compressor blades from contaminant accumulation and to avoid unstable fog formation due to a partial or complete blockage in the nozzles. Moreover, unstable fog formation could lead to water droplet formation instead of fog, which will be carried over by the air and cause severe damage to the air compressor.

In the present work, the following assumptions were made for the fogging system:

Table (11): Fogging system design values

Fogging System Design Assumptions	
Effectiveness	100%
Inlet air pressure drop	Negligible

In addition, it has been assumed that the added water in the fogging system is fully evaporated. Therefore, the water evaporation rate can be calculated as flow:

$$R = \frac{\dot{m}_{eva.water}}{\rho_{water}} \quad (36)$$

Where:

- R : Evaporation rate (m^3/s)
- ρ_{water} : Water density (kg/m^3)

Evaporator cooler

The cooling process in the evaporator cooler takes place by recirculating water over an evaporative cooling media that is located after the air intake filters. The water recirculation makes the media continuously wet, where the hot air stream will flow through it and evaporate some of the recirculating water in the media. By this, the inlet air temperature drops, and its humidity increases. Figure 25 shows a typical evaporative cooler that is used in gas turbines.

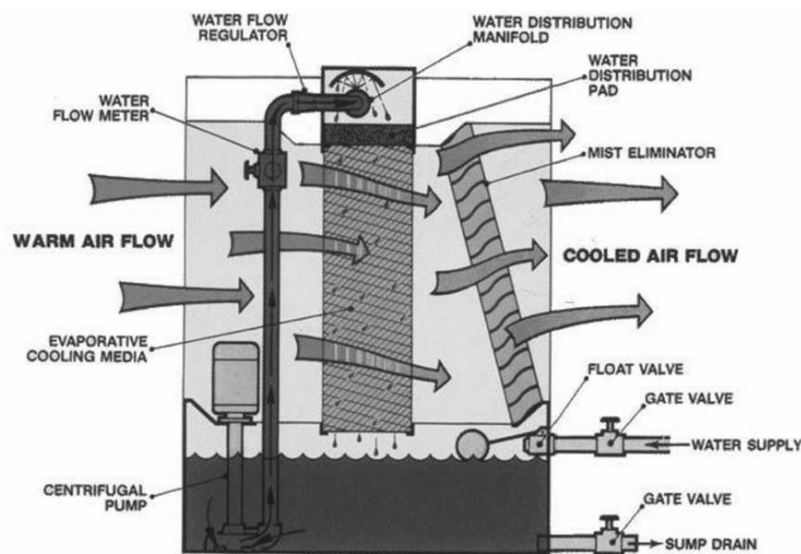


Fig. 25: Typical GT evaporative cooler: Najjar, Y. S. H., & Al-Zoghool, Y. M. A. (2015). Sustainable energy development in power generation by using green inlet-air cooling technologies with gas turbine engines. *Journal of Engineering Thermophysics*, 24(2), 181-204.

The evaporative cooler package has a mist eliminator to prevent water carryover to the air compressor. The mist eliminator, in addition to the evaporative cooling media, created more restrictions on the air stream, which increased the inlet pressure losses for the GT. The typical additional

pressure losses due to the evaporative cooler fixation in the inlet header can be considered one in WC (Jones & Jacobs, 2000).

In comparison with the fogging system, the evaporative cooler has a lesser evaporative surface area, which makes it less effective in cooling the air to its wet bulb temperature. The typical effectiveness of this cooling system is in a range of 0.85 to 0.9 (Mustapha et al., 2007).

The evaporation of the circulated water during the regular operation of the evaporative cooler increases the concentration of minerals in the water. Therefore, the minerals can accumulate on the evaporative media, causing a carryover to the compressor and reduced evaporation effectiveness (Jones & Jacobs, 2000).

A continuous blow down and makeup are done to keep the impurities concentration as minimal as possible. Moreover, to have a stable water level in the cooler, the added water to the system should equal the amount lost due to evaporation and blow down. Therefore, the amount of makeup water in the system equals the summation of evaporated and blow down rates as given by equation 37.

$$Q_{make\ up} = B + R \quad (37)$$

Where:

- $Q_{make\ up}$: The makeup water rate (m^3/s)
- B : The blow down rate (m^3/s)
- R : The evaporated water rate (m^3/s)

In addition, the evaporation rate from the evaporative cooler is calculated by equation 38, which was discussed earlier.

For the blow down rate (B), an assumption has been taken as per (Marzouk et al., 2013) and (Johnson, 1994), where the blow down rate for water that hardness is 150 ppm is given by:

$$B = 4R \quad (38)$$

Therefore, the makeup water volumetric flow rate can be written as follows:

$$Q_{make\ up} = 4R + R = 5R \quad (39)$$

In this research, the following assumptions were made for the evaporative cooler:

Table (12): Evaporative Cooler Design Values

Evaporative Cooler System Design Assumptions	
Effectiveness	90%
Additional inlet pressure losses	1 in WC

3.10.2 Refrigeration Cooling

Refrigeration cooling is the second method used to reduce the effect of high temperature at the compressor inlet of GT. Refrigeration cooling is a heat transfer process that cools the hot fluid by a heat exchanger that contains a refrigerant gas as a cold fluid.

The advantage of this method is that the wet bulb temperature does not limit the air cooling, and the air can be cooled to any desired temperature. Figure 27 illustrates the refrigeration cooling process on the psychrometric chart.

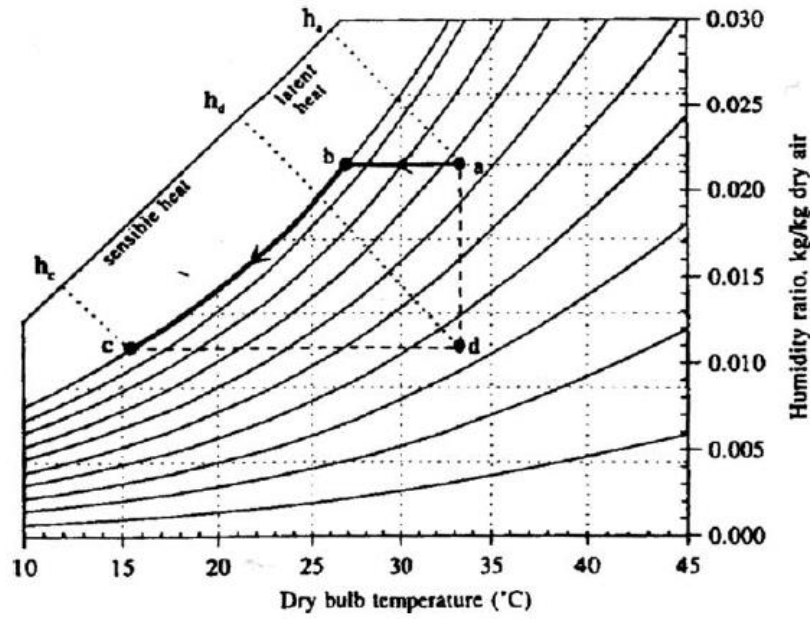


Fig. 26: Air cooling and dehumidifying process: Ameri, M., & Hejazi, S. H. (2004). The study of capacity enhancement of the Chabahar gas turbine installation using an absorption chiller. *Applied Thermal Engineering*, 24(1), 59–68. [https://doi.org/10.1016/s1359-4311\(03\)00239-4](https://doi.org/10.1016/s1359-4311(03)00239-4)

The refrigeration cooling process follows a constant specific humidity line on the psychrometric chart until the air reaches its dew point temperature. It is shown as a line from point a to point b in Figure 26. If the air outlet temperature is below the dew point, the vapor in the air stream will condense, which is called dehumidifying. The dehumidifying process follows the 100% relative humidity line, as shown in Figure 26 as a line from point b to point c.

The amount of heat that the refrigeration cooling has removed is calculated by applying the mass and energy balance equation on the system, given by equation 40 (BOONNASA et al., 2006):

$$\dot{Q}_{CL} = \dot{m}_{da}[(h_a - h_c) - h_{fg,c}(\omega_a - \omega_c)] \quad (40)$$

Where:

- $\dot{Q}_{CL}$: Cooling Load (kW)

- $\dot{m}_{da}$: Dry air mass flow rate (kg/s)
- h_a : Inlet air enthalpy (kJ/kg)
- h_c : Outlet air enthalpy (kJ/kg)
- $h_{fg,c}$: Latent heat of vaporization at outlet temperature (kJ/kg)
- ω_a : Inlet specific humidity (kg water/kg dry air)
- ω_c : Outlet specific humidity (kg water/kg dry air)

The condensate water mass balance is given by:

$$\dot{m}_{c,w} = \dot{m}_{da}(\omega_a - \omega_c) \quad (41)$$

The input power requirement to drive a refrigeration cooling system can be calculated using the coefficient of performance:

$$\text{Input Power (kW)} = \frac{\dot{Q}_c}{COP} \quad (42)$$

Mechanical chillers

The mechanical chiller is one of the refrigeration systems that is used to cool the inlet air of gas turbines. The mechanical chiller works under the vapor compression refrigeration cycle, which uses refrigerant gases as a working fluid, such as R341a and R410. Figure 27 shows a typical vapor compression cycle.

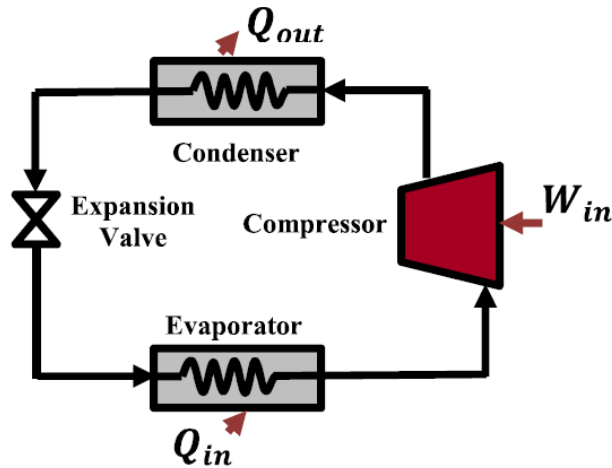


Fig. 27: Vapor compression refrigeration cycle: Kamal, S. N. O., Salim, D. A., Fouzi, M. S. M., Khai, D. T. H., & Yusof, M. K. Y. (2017). Feasibility study of turbine inlet air cooling using mechanical chillers in Malaysia climate. *Energy Procedia*, 138, 558-563.

The cooling process starts with compressing the refrigerant gas to the condenser pressure by a compressor. Then, the heat rejection process takes place in the condenser to condense the refrigerant. Next, the refrigerant gas is throttled by the expansion valve to the evaporator pressure, which causes a significant drop in its temperature. Finally, the cooling effect will take place in the evaporator, where it uses the cold environment heat to evaporate the refrigerant.

For the GT air inlet cooling application, the evaporator can be installed directly after the intake filters, or a secondary fluid, such as water, can be used to cool the inlet air stream. Figure 28 shows a schematic diagram of a gas turbine plant that uses air intake refrigeration cooling.

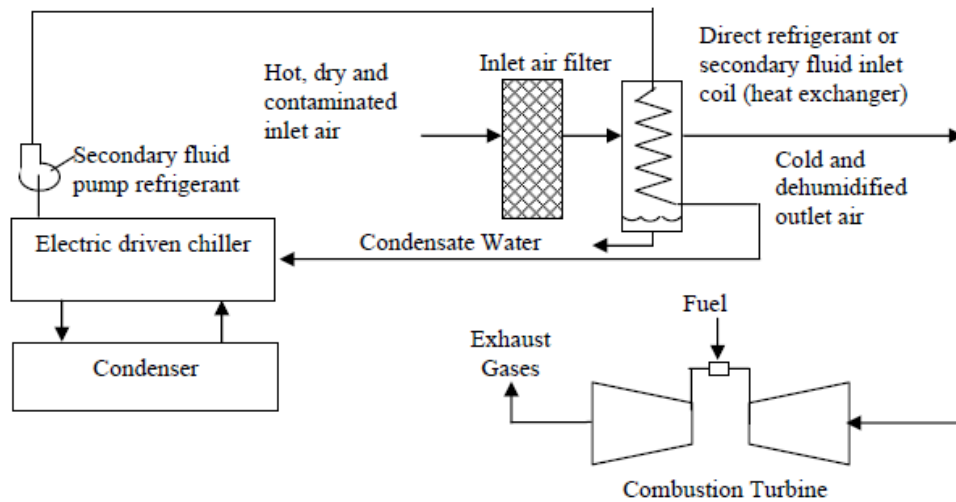


Fig. 28: Gas Turbine Plant using a refrigeration air intake cooling method: Kamal, N.A. and A.M. Zuhair, 2006. Enhancing gas turbine output through inlet air cooling. *Sudan Eng. Society J.*, 52: 1-8.

Although mechanical chillers can cool the air to any desired temperature, they have some drawbacks. Their main issue is the high power consumption by the compressor, which may make them an uneconomical cooling alternative. Moreover, a mist eliminator should be installed after the heat exchanger in the air inlet duct to prevent the condensate water from carrying over to the air compressor. This increases the air intake pressure losses, which decreases the GT performance. The typical additional pressure losses could be assumed to be 1 in WC (Jones & Jacobs, 2000).

In the present work, the following values have been chosen to model the mechanical chiller:

Table (13): Mechanical chiller design values

Mechanical Chiller System Design Assumptions	
Input Power	Electrical
COP	3.5
Inlet Air Pressure Drop	1 in WC
Desired Gas turbine air Inlet Temperature	15 °C

Absorption chiller

The absorption refrigeration cooling system is one alternative that can be used to cool the inlet air of gas turbines. Its working principle is almost identical to the vapor compression refrigeration system. The main difference is that the compressor has been replaced by an absorption mechanism that increases the pressure of the refrigerant (Çengel & Boles, 2015). Figure 29 shows a schematic diagram of the cycle.

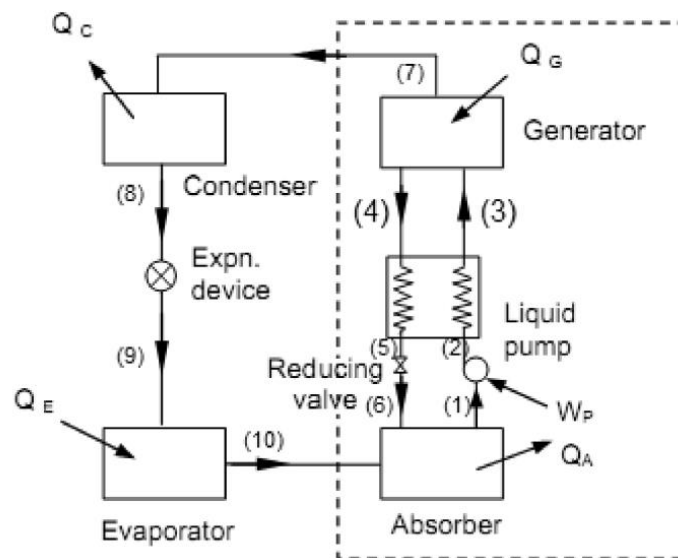


Fig. 29: Absorption refrigeration cycle: Najjar, Y. S. H., & Al-Zoghool, Y. M. A. (2015). Sustainable energy development in power generation by using green inlet-air cooling technologies with gas turbine engines. *Journal of Engineering Thermophysics*, 24(2), 181-204.

The absorption refrigeration cycle usually uses ammonia and water as working fluids, where the ammonia is the refrigerant and the water is a transport medium only.

In the absorber, the low-pressure ammonia vapor will dissolve in the water to form a solution. This reaction releases heat, which increases the absorber temperature. The ability of ammonia absorption is inversely proportional to the temperature. Therefore, the absorber should be continuously cooled to ensure a maximum amount of absorption (Çengel & Boles, 2015).

The ammonia water solution that is rich with ammonia is transferred from the absorber to the generator by a feed pump, which increases the solution pressure to the required condenser pressure. In the generator, a heat addition process takes place by an external source to separate the ammonia from the water, which occurs due to the boiling point difference. The high-pressurized ammonia gas will follow the vapor compression cycle steps, which were discussed earlier. It will be condensed in the condenser, throttled to the evaporator pressure, and finally, evaporated in the evaporator.

Moreover, to increase the COP of the cycle, the heat of the hot solution that is transferred from the generator to the absorber will be utilized to increase the temperature of the cold solution that has been pumped to the generator. Therefore, the cycle will require less heat input for the same operating condition.

It is worth mentioning that water and lithium bromide have also been used in absorption refrigeration cycles as an environment-friendly alternative. In this case, the water is the refrigerant, and the lithium bromide is the transport medium.

The thing that distinguishes between mechanical and absorption chillers is that in the absorption chiller, the refrigerant pressure is increased by a pump, which requires relatively less power input compared to a compressor (Çengel & Boles, 2015). Moreover, the absorption chiller can be operated on a low-quality heat source that ranges from 100 C to 200 C only (Hiwa Khaledi et al., 2005).

The absorption chiller follows a similar manner as the mechanical chiller to cool the inlet air of the GT, which is shown in Figure 30.

Two heat sources were adopted to drive the absorption chiller. The first one is a low-pressure super-heated steam extracted from the LP steam header of the CCGT plant. In this option, the steam heat will be utilized until the steam condenses fully in the generator of the chiller. Then, the saturated liquid will be sent back to the cycle before the deaerator. Figure 30 illustrates the process.

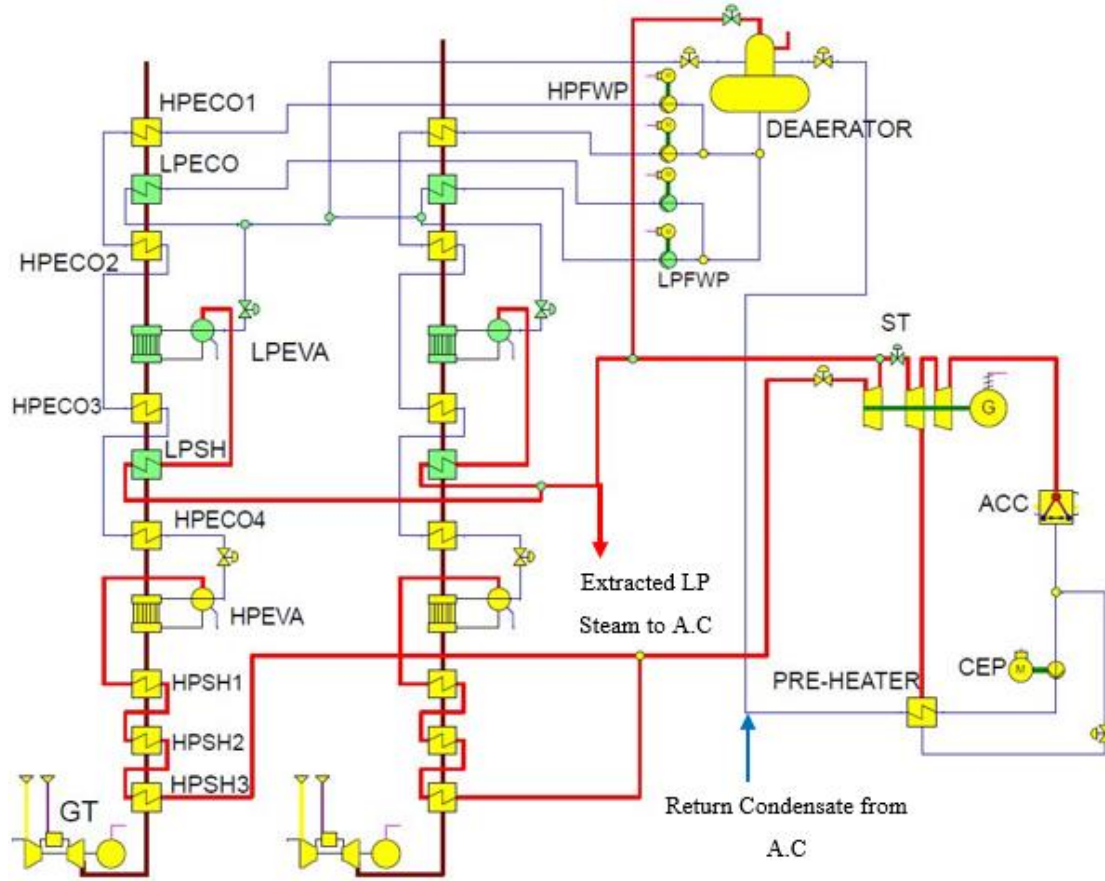


Fig. 30: Plant layout utilizing a low pressure super-heated steam source to drive the absorption chiller

The extracted steam mass flow rate depends on the required heat by the generator and it can be calculated by:

$$\dot{m}_{ext} = \frac{\dot{Q}_G}{h_{s,in} - h_{f,out}} \quad (43)$$

Where:

- $\dot{m}_{ext}$: Extracted steam mass flow rate (kg/s)

- $\dot{Q}_G$: Generator heat requirement (kW)
- $h_{s,in}$: LP super-heated steam enthalpy (kJ/kg)
- $h_{f,out}$: saturated liquid water enthalpy (kJ/kg)

The second heat source to drive the absorption chiller is a waste heat source. Table 14 contains the design values that have been chosen to model the absorption chiller.

Table (14): Absorption Chiller Design Values

Absorption chiller design values	
Input heat source	Internal : Steam at P: 4.9 Bar and T: 265 °C
	Waste heat
COP	1
Inlet Air Pressure Drop	1 in WC

3.10.3 Comparison between inlet air cooling systems

This section will illustrate the advantages and disadvantages of each cooling alternative used in the present work. Table 15 shows the comparison (Deng et al., 2020), (Al-Ibrahim & Varnham, 2010).

Table (15): Air Intake Cooling Systems Comparison

Cooling System	Advantages	Disadvantages
Fogging	1- Low power requirements 2- Low initial, operation, and maintenance cost 3- High effectiveness 4- No increase in GT intake pressure losses	1- Cooling Restrictions. 2- Requires demineralized water
Evaporative Cooler	1- Low power requirements 2- Low initial, operation, and maintenance cost 3- Uses less quality water compared to the fogging system	1- Cooling Restrictions ambient wet bulb temperature. 2- Lower effectiveness compared to fogging. 3- Increasing GT intake pressure losses. 4- Continuous water blow downs to maintain the circulated water quality
Mechanical chiller	1- The cooling is not limited by ambient conditions. 2- The GT air intake temperature can be a fixed value. 3-High COP	1- High initial, operation, and maintenance costs 2- High Power Consumption 3- Increasing the GT intake pressure losses.
Absorption Chiller	1- The cooling is not limited by ambient conditions. 2- The GT air intake temperature can be a fixed value. 3- Minor electricity consumption compared to mechanical chillers. 4- Waste heat can be used to drive the chiller.	1- High initial, operation, and maintenance costs. 2- Low COP. 3- Increasing the GT intake pressure losses.

3.11 The Selection Process

In the present work, the selection process among the GT air intake cooling systems is based on technical and economic analysis. Six different cases have been created as follows:

1. CCGT plant without cooling (Base Case).
2. CCGT plant with fogging cooling system.
3. CCGT plant with evaporative cooler system.
4. CCGT plant with mechanical chiller.
5. CCGT plant with internal steam absorption chiller.
6. CCGT plant with a waste heat absorption chiller.

The importance of the technical analysis is to provide the amount of additional power generation and fuel consumption associated with each cooling system that has been used. The results of the technical analysis are considered the primary input to the economic analysis, aligned with all other costs and revenues accumulated as a result of adding an air intake cooling system to the plant.

3.11.1 Technical Analysis

The first step in the technical analysis is determining the required design cooling capacity and the resources needed to drive the cooling systems in the selected CCGT. Therefore, the model should be first simulated for the highest hourly average ambient temperature in Bahrain. As per the collected data for 2022, the hottest day in Bahrain was the 17th of August, where the maximum hourly average ambient temperature reaches around 48 °C. Figure 31 shows the climate data during the peak hours of the mentioned date, which have been extracted from the weather station of the selected CCGT plant.

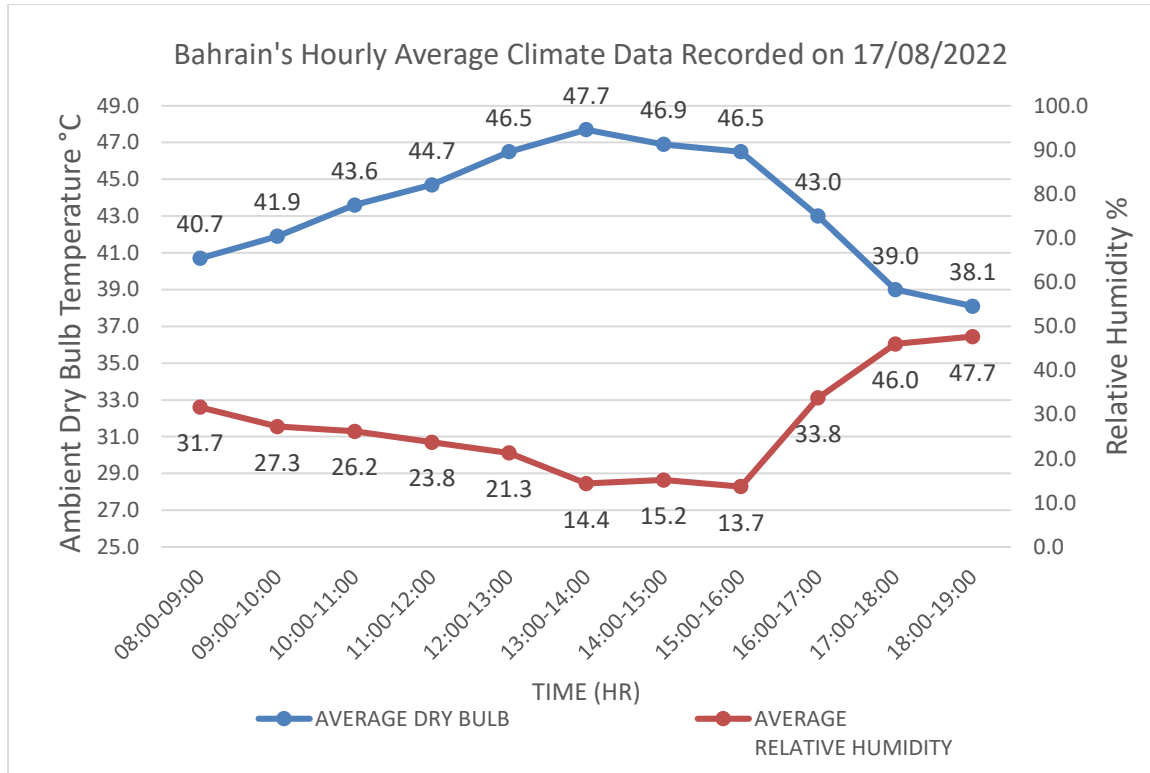


Fig. 31: Bahrain's climate data on 17th of August, 2022

Next, to determine the yearly performance behavior of each case, Bahrain's monthly average climate data have been considered as input variables. The simulation result of the base model represents the baseline performance of the plant, which has been used to determine the augmented power and extra fuel consumption by the remaining cases. Figure 32 shows the monthly average ambient temperature and relative humidity for 2022.

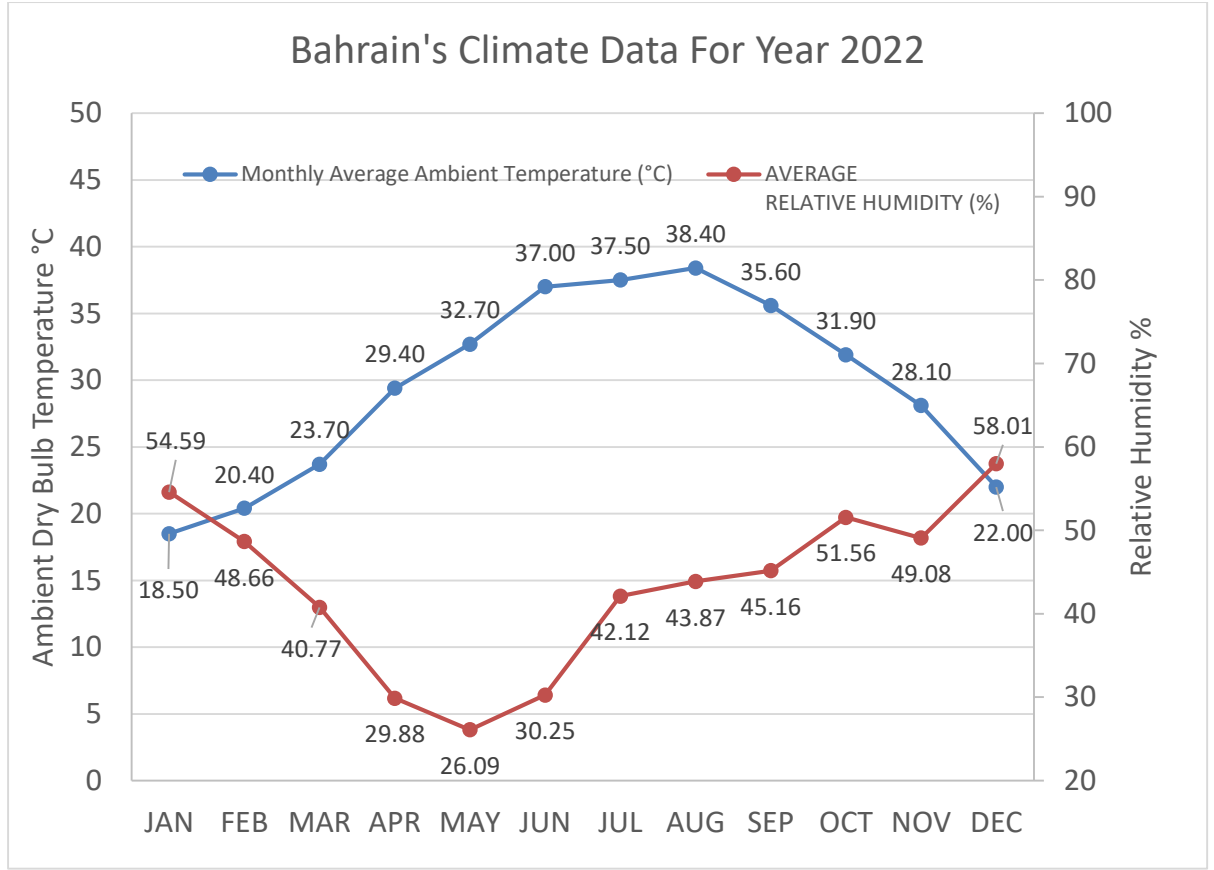


Fig. 32: Bahrain's monthly climate data for the year 2022

3.11.2 Economic Analysis

The basis of any project selection is the economic justification. Therefore, an economic evaluation is required to select the most economically justified alternative for Bahrain's operating conditions. For the present work, the following economic analysis tools were selected to evaluate the cooling alternatives (Barigozzi et al., 2015), (Saeed et al., 2011):

1. Net present value (NPV)
2. Internal rate of return (IRR)
3. Payback period (PB)

The net present value is given by:

$$NPV = \sum_{n=0}^N \frac{C_n}{(1+r)^n} \quad (44)$$

Where:

- C_n : Net cash flow
- r : Interest rate
- n : Number of years

The internal rate of return is calculated by solving for r as follows:

$$NPV = \sum_{n=0}^N \frac{C_n}{(1+r)^n} = 0 \quad (45)$$

Where:

- N = Payback period

The Payback period is given by:

$$Payback = \frac{\text{Initial investment}}{\text{Annual cash flow}} \quad (46)$$

The cost of a GT inlet air cooling system is computed by considering the following costs:

Installation cost (C_i): For the present work, and by referring to the open literature, typical installation cost values have been considered based on \$ per each additional generated kW by the GT. The fogging system installation cost has been considered as \$ 45 /kW (Seyed et al., 2014). Moreover, the evaporative cooler system has more parts and requires more modifications to the air inlet duct and more installation time (Jones & Jacobs, 2000). Therefore, its installation cost has been considered \$ 50/kW (Marzouk et al., 2013).

For refrigeration cooling systems, the typical initial cost of an absorption chiller is considered more expensive than the mechanical chiller at the same capacity (Santos & Andrade, 2012). Therefore, the installation cost of the absorption and mechanical chillers are considered \$ 300 /kW and \$ 250 /kW, respectively (Seyed et al., 2014).

In addition, the sizing of a cooling system is done based on the maximum cooling load required, which occurs at the highest ambient temperature (48 °C for Bahrain). Thus, the installation cost of a cooling system is given by equation 49.

$$C_i = R_i * P_{GT, ADDED} \quad (47)$$

Where:

- C_i : Cooling system's installation cost (\$)
- R_i : The rate of cost per each augmented kilowatt (\$/kW)
- $P_{GT, ADDED}$: The additional power generation by the GT (kW)

Annual operation and maintenance cost ($C_{O\&M}$): For the annual operation and maintenance cost, the fogging system has fewer parts, and it is easier to operate and maintain compared to the evaporative cooler. Therefore, the annual operation and maintenance cost for the fogging system and evaporative cooler has been considered to be 3% and 4% of the installation cost, respectively (Marzouk et al., 2013), (Hosseini et al., 2007).

For the absorption and mechanical chillers, the operation and maintenance costs have been considered as 6% (Marzouk et al., 2013).

Annual additional fuel cost (C_f): The additional fuel costs accumulating due to the additional power generation are computed based on a gas price of \$ 4 / MBTU.

Annual water consumption cost (C_w): The main requirement of evaporative cooling systems is water. The fogging system uses demineralized water, whereas less quality water can be supplied to the evaporative cooler. The water cost for the fogging system has been considered as \$ 3 / m³. On the other hand, the water cost for the evaporative cooler has been considered as \$ 2 / m³ as per Bahrain's electricity and water authority's latest tariffs for the non-domestic sector (EWA, 2023).

The revenues of the cooling systems are generated by selling the generated energy. For the present work, the electricity tariff has been considered as \$ 0.08 / kWh (29 Fills /kWh) as per the electricity and water authority in Bahrain (EWA, 2023). The economic analysis is based on equal live projects of 20 years with an interest rate of 6%. Tables 16 and 17 summarize the economic analysis input values and cooling system costs, respectively.

Table (16): Economic analysis input values

Parameter	Value
Interest Rate (%)	6%
N (Years)	20
Electricity Price (\$/kWh)	0.08
Gas Cost (\$/ MBTU)	4

Table (17): Cooling systems costs

Cooling System	Installation Cost (\$/kW Added)	Annual O&M (%)	Water cost (\$/m ³)
Fogging	45	3%	3
Evaporative Cooler	50	4%	2
Mechanical chiller	250	6%	Neglected
Absorption Chiller	300	6%	Neglected

Chapter Four

Results and Discussion

4.1 Introduction

4.2 Model Validation

4.3 Parametric Study

4.4 Models Simulations Results

4.5 Technical Analysis

4.6 Economic Analysis

4.7 The Selection

CHAPTER 4: Results and Discussion

4.1 Introduction

This chapter will present the simulation results of the selected cases using the climate data of Bahrain. Moreover, a parametric study will take place to study and quantify the effect of the ambient temperature on the CCGT plant, which is one of the main objectives of this research. Appendix A shows the calculation flow charts of the used models.

4.2 Model validation

To ensure the accuracy of the build model, validation has been done by comparing the simulation results with the manufacturer's data. The validation was done at two different ambient temperatures to ensure the maximum accuracy of the design. Tables 18 and 19 show the validation results at 15 °C and 42 °C, respectively. In addition, Appendix B shows the same results in Epsilon.

Table (18): Model Validation at 15°C

Ambient Conditions (Temperature, RH, Pressure)	(15 °C, 60% , 1.013 bar)		
Gas Turbine :	Manufacturer	Model	Error %
Power Output (MW)	181.06	180.94	-0.07%
Inlet Fuel Mass Flow Rate (kg/s)	14.21	14.20	-0.03%
Exhaust Gas Temperature (°C)	506.40	506.36	-0.01%
Exhaust Gas Mass Flow Rate (kg/s)	547.60	547.58	0.00%
Heat Recovery Steam Generator :	Manufacturer	Model	Error %
HP Steam Outlet Mass Flow Rate (kg/s)	60.50	59.99	-0.85%
HP Steam Outlet Temperature (°C)	490.80	491.20	0.08%
LP Steam Outlet Mass Flow Rate (kg/s)	15.70	15.98	1.78%
LP Steam Outlet Temperature (°C)	269.60	265.42	-1.55%
Exhaust Gas Temperature (°C)	154.50	156.58	1.35%
Steam Turbine :	Manufacturer	Model	Error %
HP Steam Inlet Flow (kg/s)	121.00	119.98	-0.85%
HP Steam Inlet Temperature (°C)	489.90	491.20	0.27%
LP Steam Inlet Flow (kg/s)	22.10	22.47	1.69%
LP Steam Inlet Temperature (°C)	269.60	265.42	-1.55%
Extracted LP Steam Mass Flow Rate To Deaerator (kg/s)	9.20	9.49	3.11%
Extracted Steam Mass Flow Rate To Pre-Heater (kg/s)	9.60	9.00	-6.30%
Extracted Steam Temperature To Pre-Heater (°C)	94.30	92.03	-2.41%
Exhaust Steam Mass Flow Rate (kg/S)	133.57	133.45	-0.09%
Exhaust Steam Pressure (mbar)	140.00	140.00	0.00%
Exhaust Steam Temperature (°C)	52.55	52.55	0.00%
Power Output (MW)	129.14	129.91	0.60%
Total Plant (2 GT, 2 HRSG, 1 ST):	Manufacturer	Model	Error %
Plant Power Output (MW)	491.27	491.78	0.10%
Plant LHV Efficiency (%)	50.46	50.53	0.13%

Table (19): Model Validation at 42 °C

Ambient Conditions (Temperature, RH, Pressure)	(42 , 60%, 1.013 Bar)		
Gas Turbine :	Manufacturer	Model	Error %
Power Output (MW)	153.92	153.62	-0.20%
Inlet Fuel Mass Flow Rate (kg/s)	12.65	12.66	0.12%
Exhaust Gas Temperature (°C)	524.10	524.71	0.12%
Exhaust Gas Mass Flow Rate (kg/s)	497.40	496.84	-0.11%
Heat Recovery Steam Generator :	Manufacturer	Model	Error %
HP Steam Outlet Mass Flow Rate (kg/s)	59.50	59.22	-0.46%
HP Steam Outlet Temperature (°C)	506.80	506.77	-0.01%
LP Steam Outlet Mass Flow Rate (kg/s)	14.10	14.27	1.21%
LP Steam Outlet Temperature (°C)	269.20	266.20	-1.11%
Exhaust Gas Temperature (°C)	153.10	155.27	1.42%
Steam Turbine :	Manufacturer	Model	Error %
HP Steam Inlet Flow (Kg/s)	118.90	118.45	-0.38%
HP Steam Inlet Temperature (°C)	506.00	506.77	0.15%
LP Steam Inlet Flow (kg/s)	19.80	19.96	0.79%
LP Steam Inlet Temperature (°C)	269.20	266.20	-1.11%
Extracted LP Steam Mass Flow Rate To Deaerator (kg/s)	8.30	8.58	3.41%
Extracted Steam Mass Flow Rate To Pre-Heater (kg/s)	5.60	5.44	-2.87%
Extracted Steam Temperature To Pre-Heater (°C)	94.30	93.92	-0.40%
Exhaust Steam Mass Flow Rate (kg/s)	133.15	132.97	-0.14%
Exhaust Steam Pressure (mbar)	300.20	299.77	-0.14%
Exhaust Steam Temperature (°C)	70.16	69.08	-1.54%
Power Output (MW)	124.45	123.72	-0.58%
Total Plant (2 GT, 2 HRSG, 1 ST):	Manufacturer	Model	Error %
Plant Power Output (MW)	432.30	430.97	-0.31%
Plant LHV Efficiency (%)	49.87	49.66	-0.43%

All presented values have an acceptable variation. Therefore, Ebsilon could be considered an accurate and powerful tool for power plant modeling and simulation.

4.3 Parametric study

The build model of the CCGT plant has been simulated at ambient temperatures ranging from 15 °C to 45 °C to study the effect of the compressor intake temperature on the plant performance. Figures from 33 to 42 show the parametric study results.

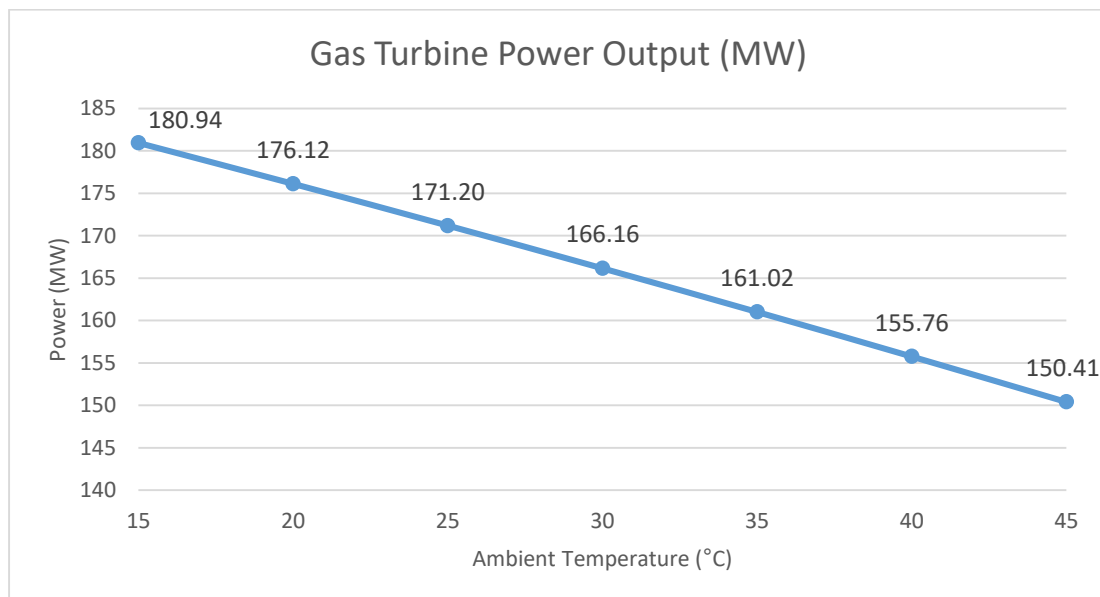


Fig. 33: Effect of ambient temperature on GT power output

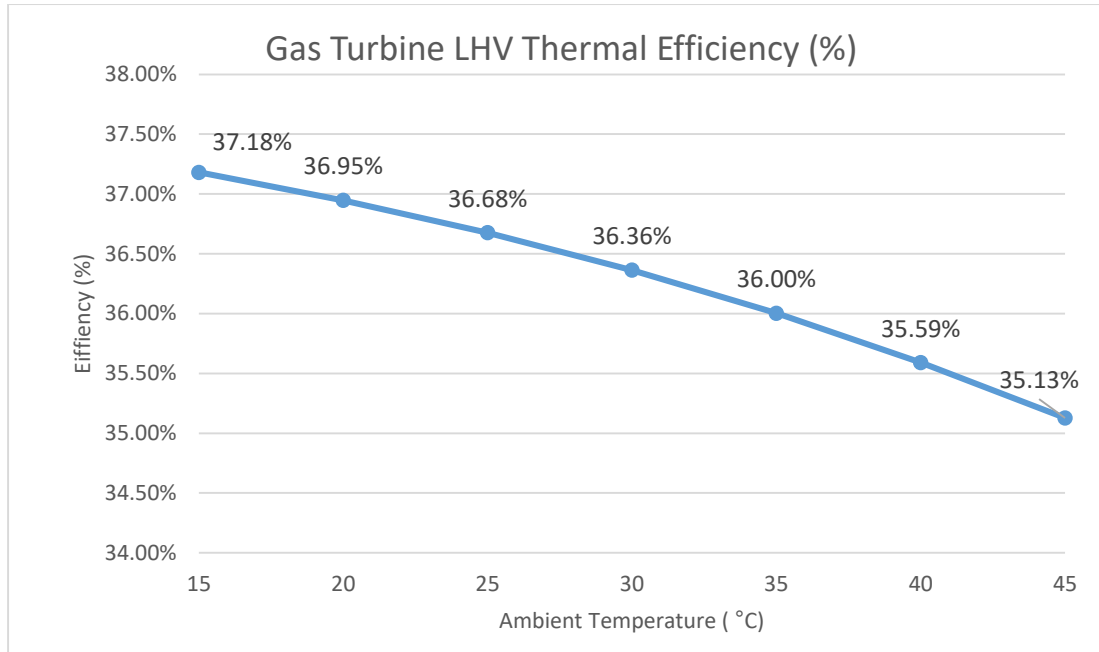


Fig. 34: Effect of ambient temperature on GT thermal efficiency

Figure 33 shows that the GT power output is inversely proportional to the ambient temperature, where the GT power output decreased by 17% as the ambient temperature increased from 15 °C to 45 °C. Moreover, Figure 34 shows that the thermal efficiency of the gas turbine also decreases with the increase of the ambient temperature. The GT efficiency drops by 5.5% as the ambient temperature increases from 15 °C to 45 °C.

The reason behind this is that as the ambient temperature increases, the inlet air mass flow rate decreases due to the decrease in the air density. In addition, the air compressor requires more work to compress hotter air, as discussed earlier.

The exhaust flue gas condition from the gas turbine has a critical influence on the water-steam cycle since it is the heat source that drives the ST. Figures 35 and 36 show the effect of the ambient temperature on the GT exhaust mass flow rate and temperature, respectively.

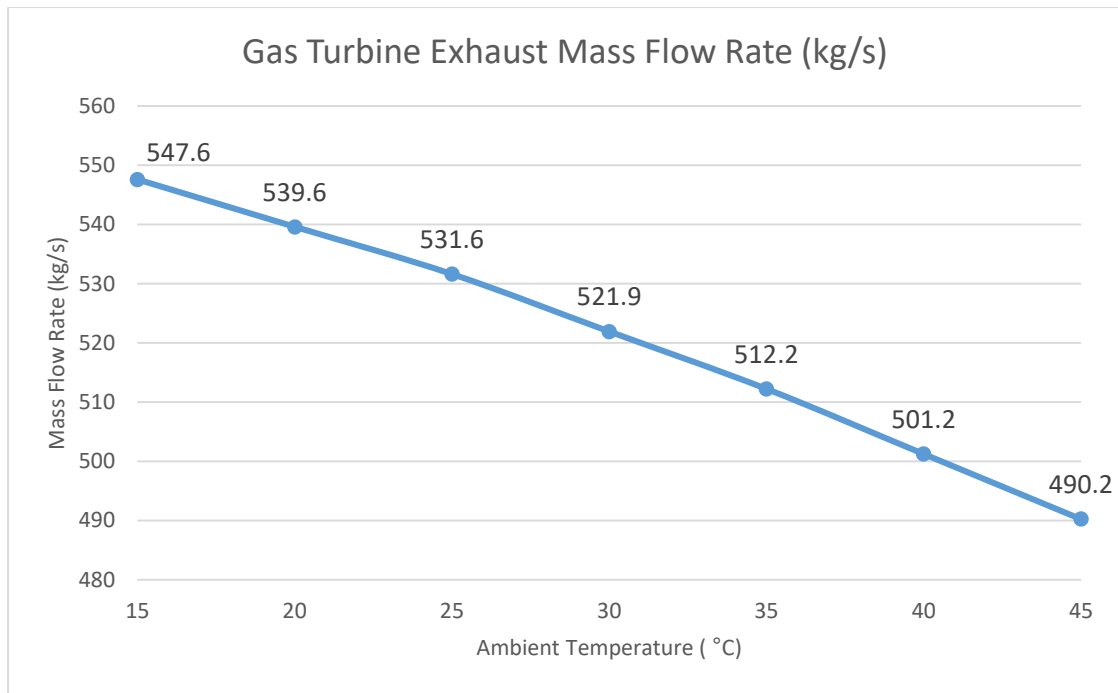


Fig. 35: Effect of ambient temperature on GT exhaust mass flow rate

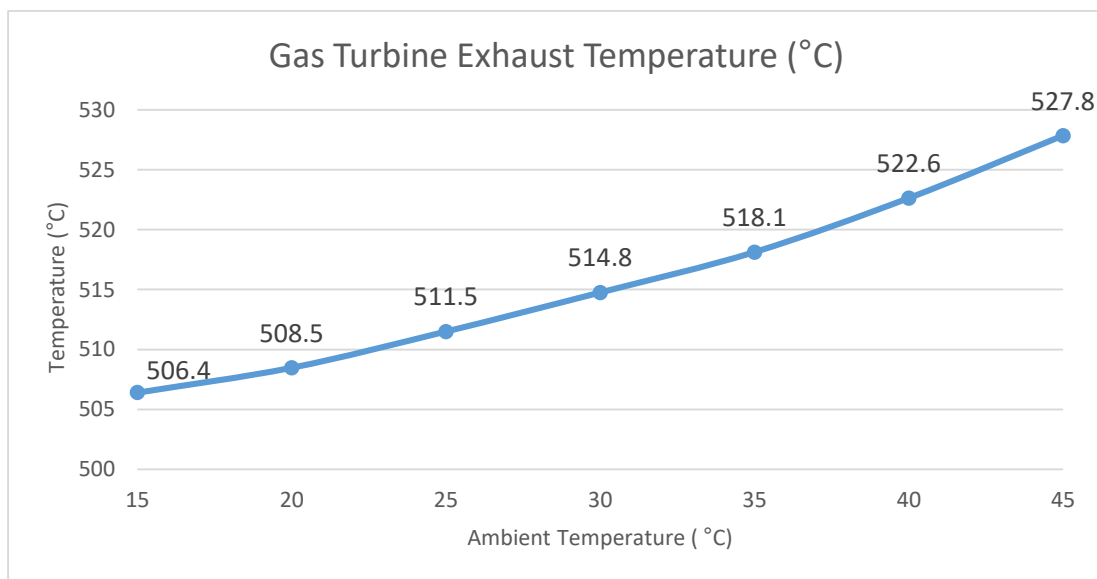


Fig. 36: Effect of ambient temperature on GT exhaust temperature

It can be seen from the figures that increasing the ambient temperature decreases the exhaust gas mass flow rate. This is due to the lower inlet air and fuel mass flow rates at higher ambient temperatures.

The temperature after the turbine is directly proportional to the ambient temperature because the pressure ratio is inversely proportional to the

ambient temperature, and the temperature after the turbine is inversely proportional to the pressure ratio, as discussed earlier.

This increase in the GT exhaust temperature limits the impact on the HRSG. Figure 37 shows the effect of ambient temperature on the generated HP steam mass flow rate. As the ambient increases from 15 °C to 45 °C, the generated steam decreases by 1.1 % only. Thus, the ambient temperature has a marginal effect on the produced steam in the HRSG (Salim et al., 2020).

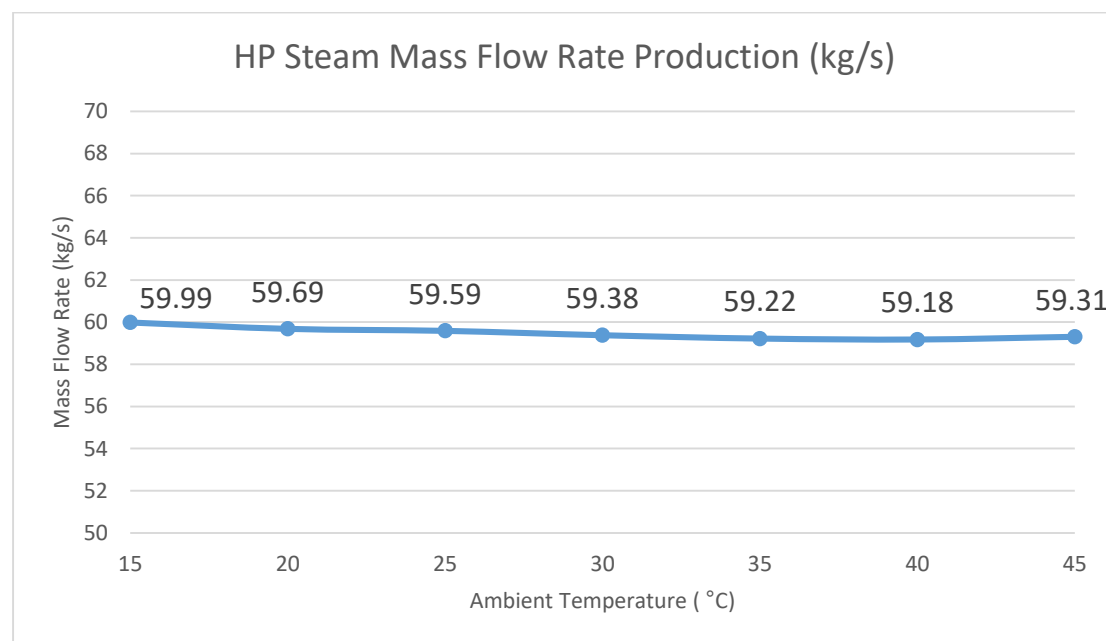


Fig. 37: Effect of ambient temperature on the HP steam production

Figure 38 shows that the generated steam temperature is directly proportional to the ambient temperature because the increased ambient temperature increases the exhaust gas temperature, which increases the generated steam temperature in the HRSG.

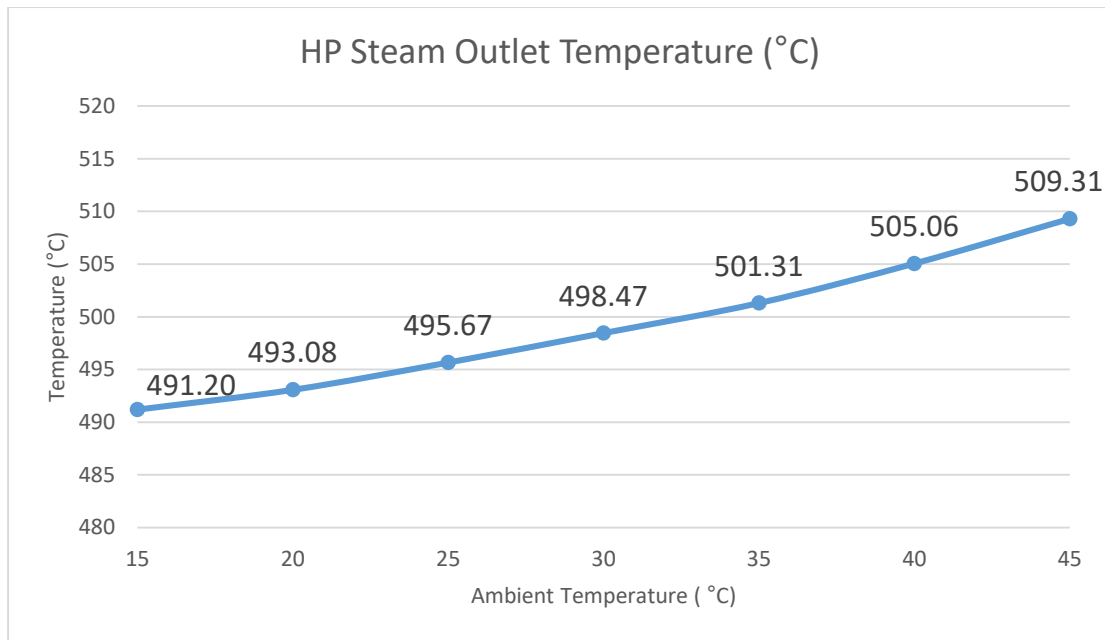


Fig. 38: Effect of ambient temperature on the HP steam outlet temperature

Figure 39 shows the effect of the ambient temperature on the condenser pressure. It can be seen that the ACC can maintain the pressure at its set point (140 mbar) until the air temperature reaches around 25 °C. As the ambient increases, the pressure will also increase due to the lower inlet air mass flow rate to the axial flow fans at higher ambient temperatures.

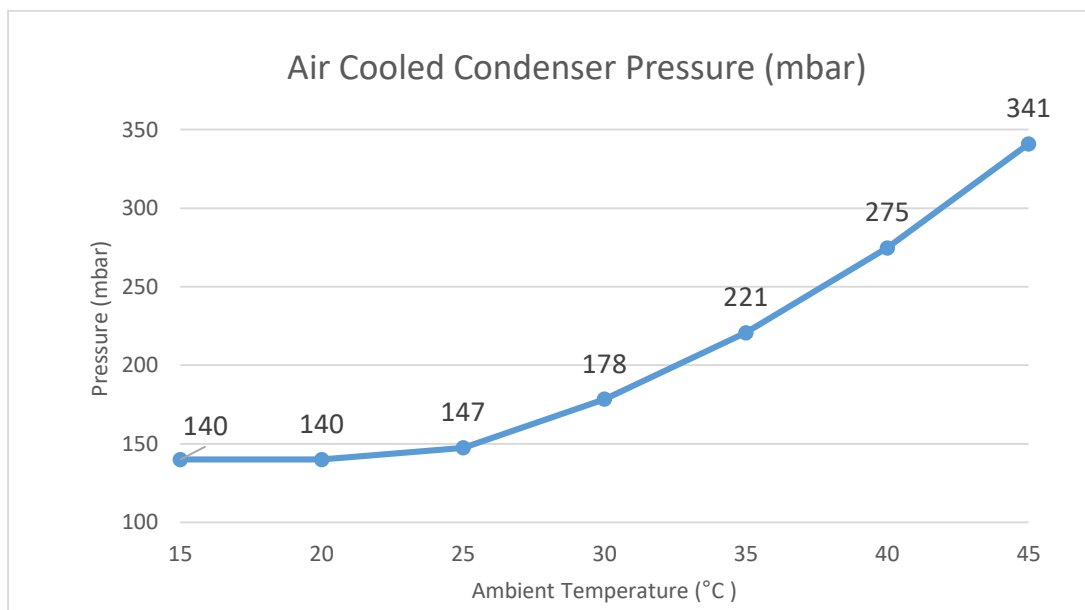


Fig. 39: Effect of ambient temperature on the condenser pressure

Figure 40 shows the effect of the ambient temperature on the ST output. The ST has less sensitivity to the ambient temperature than the GT, where its output decreases by 6% only when the temperature increases from 15 °C to 45 °C.

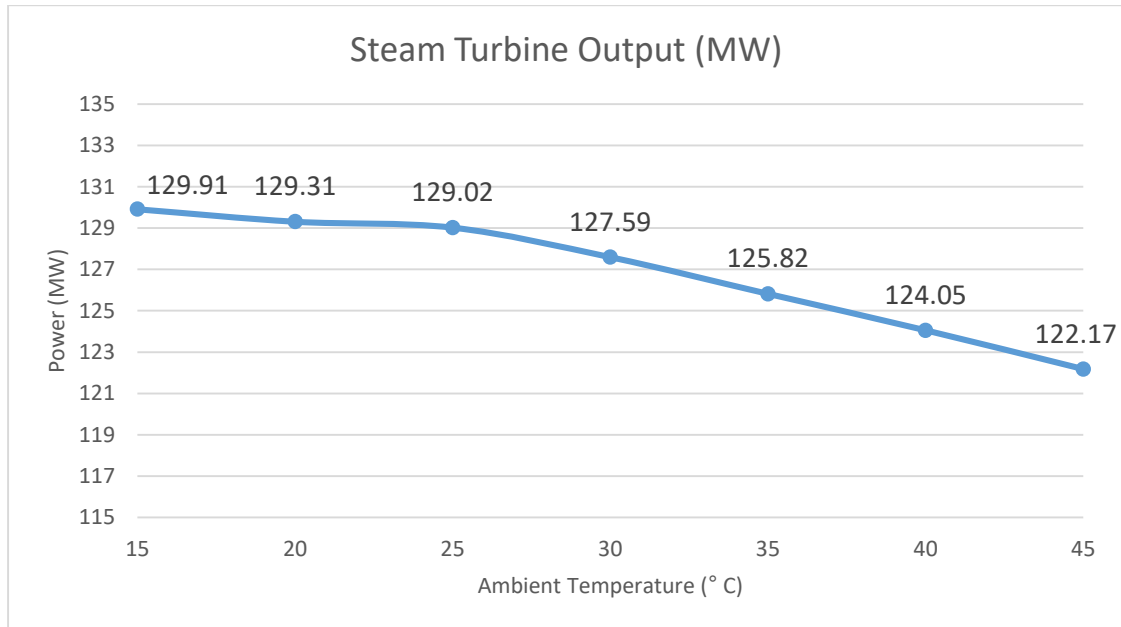


Fig. 40: Effect of ambient temperature on the steam turbine power output

Moreover, the ST output decreased by one Mw only in the range of 15 °C to 25 °C while the reduction rate increased after 25 °C, due to the increase of the condenser pressure, which decreases the expansion ratio of the steam.

The overall plant output follows the same behavior as the GT, as seen in Figure 41. The total plant output decreases by 14 % as the ambient increases from 15 °C to 45 °C.

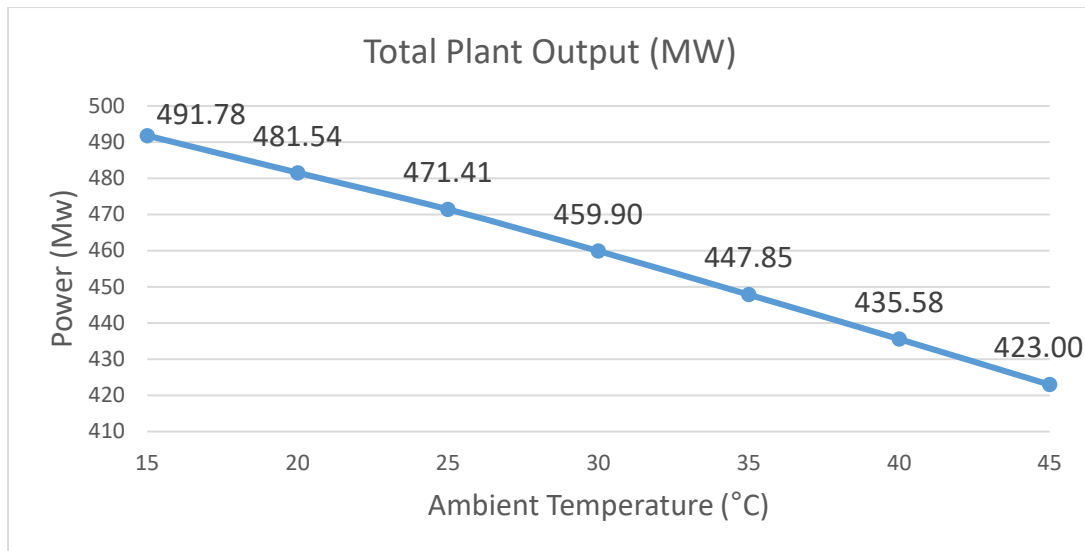


Fig. 41: Effect of ambient temperature on the total plant power output

The combined cycle thermal efficiency reduces with the increase in ambient temperature, as seen in Figure 42. The rate of efficiency reduction increases as the ambient temperature crosses 25 °C, where the condenser pressure starts increasing, as discussed earlier. The combined cycle plant efficiency shows less sensitivity to the ambient temperature than GT, where the combined cycle efficiency reduces by 2.2% only, while the GT efficiency reduces by 5.5% as the ambient increases from 15 °C to 45 °C.

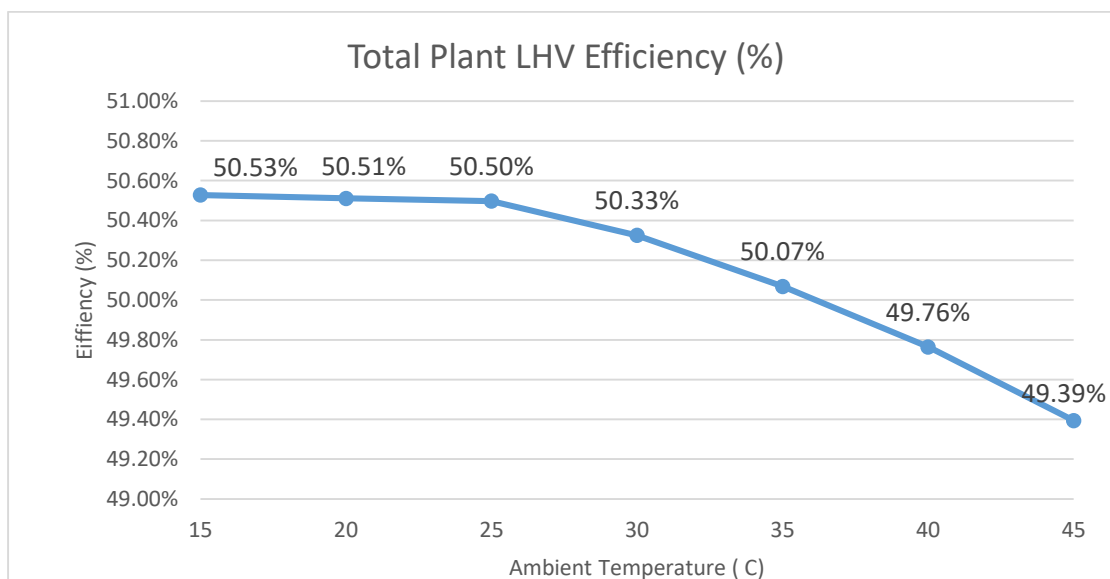


Fig. 42: Effect of ambient temperature on the total plant efficiency

The performed parametric study has confirmed that the increasing of ambient temperature affects the performance of the combined cycle plants performance, which is mainly due to the:

- The decrease in air density
- The increase in compressor work
- The increase in condenser pressure

4.4 Models Simulation Results

4.4.1 Simulation at maximum ambient temperature

The presented cases have been simulated at the maximum hourly ambient temperature recorded in Bahrain during 2022. This was done to determine the maximum cooling capacity required by the refrigeration cooling systems and to ensure the availability of sufficient water supplies for the evaporative cooling systems.

Tables 20 and 21 summarize the results of the base case and evaporative cooling systems cases.

Table (20): Evaporative Cooling Systems Results Summary at 48 °C

Model	A.T (°C)	R.H (%)	C.I.T (°C)	C.I RH %	Evaporation Rate m ³ /h	Total Water Consumption m ³ /h
Base Case	47.7	14.4	47.7	14.4	-	-
Fogging	47.7	14.4	24.5	100. 0	34	34
Evaporative Cooler	47.7	14.4	26.8	81.3	29	145.1

Table (21): Evaporative Cooling Systems Results Summary at 48 °C (Continue)

Model	GT Output (MW)	Total Plant Output (MW)	Plant HHV Efficiency (%)	Total Plant Gas Mass Flow Rate (kg/s)
Base Case	146.1	407.3	44.23%	24.23
Fogging	172.5	465.8	44.62%	27.47
Evaporative Cooler	169.2	458.5	44.57%	27.07

The simulation results show that the fogging system reduced the compressor intake temperature more than the evaporative cooler because of the effectiveness difference between both systems.

Moreover, the fogging systems show a lower water consumption than the evaporative cooler due to the need to drain the circulated water in the evaporative cooler, as discussed earlier.

Tables 22 and 23 summarize the results of the refrigeration cooling systems cases.

Table (22): Refrigeration Cooling Systems Results Summary at 48 °C

Model	A.T (°C)	R.H (%)	C.I.T (°C)	C.I RH %	Cooling Load Per GT (MW)	Cooling Load Per GT (TR)	Required Power/ Heat by the chiller Per GT (MW)
Mechanical Chiller	47.7	14.4	15	92.7	16.1	4575	4.6
Internal Steam A.C	47.7	14.4	15	92.7	16.1	4575	16.1
Waste heat A.C	47.7	14.4	15	92.7	16.1	4575	16.1

**Table (23): Refrigeration Cooling Systems Results Summary at 48 °C
(Continue)**

Model	Extracted LP Steam to Absorption Chiller Per GT (kg/s)	Total Plant Net Output (MW)	Total Plant HHV Efficiency (%)	Total Plant Gas Flow (kg/s)
Mechanical Chiller	-	472.78	43.77%	28.42
Internal Steam A.C	6.8	476.65	44.13%	28.42
Waste heat A.C	-	481.98	44.62%	28.42

The results have shown that during the peak hour, the plant with waste heat absorption chillers generated the highest amount of power. However, the effect of using an internal steam source could be seen in the internal steam absorption chiller case, where the total plant power and efficiency are reduced by around 1.11%.

In addition, the maximum demand of the steam by the absorption chiller per GT is 13.6 kg/s at an inlet pressure and temperature of 4.9 Bar and 265 °C, respectively.

The mechanical chiller case generates the lowest amount of power because of the high electrical power consumption, where the total plant needs a power of 9.2 Mw at Peak hour, which reduces the plant efficiency to 43.77%.

4.4.2 Simulations at Bahrain's Climate Data

CCGT without cooling (Base Case)

The base case model has been simulated for the Bahrain climate data. Table 24 shows the simulation results, which form the baseline of this research.

Table (24): Base Model Results Summary

Month	A.T (°C)	R.H (%)	Total Plant Output (MW)	Total Generated Energy (MWh)	Plant HHV Efficiency (%)	Total Plant Gas Flow (kg/s)	Total Inlet Gas Energy (GBTU)
Jan	18.5	54.6	484.3	360,299	45.54%	27.98	2,699
Feb	20.4	48.7	480.0	322,550	45.54%	27.74	2,417
Mar	23.7	40.8	472.7	351,707	45.54%	27.31	2,635
Apr	29.4	29.9	458.4	330,062	45.38%	26.58	2,482
May	32.7	26.1	449.7	334,604	45.24%	26.16	2,524
Jun	37.0	30.3	439.2	316,200	45.03%	25.66	2,396
Jul	37.5	42.1	439.5	326,955	45.01%	25.69	2,479
Aug	38.4	43.9	437.4	325,424	44.96%	25.60	2,470
Sep	35.6	45.2	444.6	320,126	45.11%	25.94	2,421
Oct	31.9	51.6	454.5	338,135	45.28%	26.41	2,548
Nov	28.1	49.1	463.3	333,604	45.43%	26.84	2,506
Dec	22.0	58.0	477.4	355,218	45.54%	27.59	2,661
Total	-	-	458.3	4,014,884	45.31%	-	30,237

CCGT Plant with Inlet Fogging System Case

The fogging system is one of the selected evaporative cooling systems, which is considered to be more effective than the evaporative cooler. In addition, it does not cause any inlet pressure losses since no additional parts are added to the air inlet stream. However, the fogging system must be supplied with demineralized water, which costs more than normal quality water. Tables 25 and 26 show the simulation results of the fogging system case.

Table (25): CCGT with Inlet Fogging Cooling System Results Summary

Month	A.T (°C)	R.H (%)	C.I.T (°C)	C.I RH%	Total Plant Evaporated Water Consumption (m ³)
Jan	18.5	54.6	13.1	100	6,277
Feb	20.4	48.7	13.9		6,836
Mar	23.7	40.8	15.3		9,702
Apr	29.4	29.9	17.5		13,136
May	32.7	26.1	18.9		15,580
Jun	37.0	30.3	23.0		15,238
Jul	37.5	42.1	26.4		12,593
Aug	38.4	43.9	27.5		12,313
Sep	35.6	45.2	25.5		11,052
Oct	31.9	51.6	23.8		8,962
Nov	28.1	49.1	20.2		6,830
Dec	22.0	58.0	16.6		6,326
Total	-	-	-	-	124,843

Table (26): CCGT with Inlet Fogging Cooling System Results Summary (Continue)

Month	Total Plant Output (MW)	Total Generated Energy (MWh)	Plant HHV Efficiency (%)	Total Plant Gas Flow (kg/s)	Total Inlet Gas Energy (GBTU)	Additional Generated Energy (MWh)	Added Inlet Gas Energy (GBTU)
Jan	497.4	370,030	45.56%	28.73	2,771	9,731	72
Feb	495.9	333,250	45.56%	28.64	2,496	10,700	79
Mar	493.1	366,882	45.56%	28.48	2,748	15,175	113
Apr	487.6	351,107	45.43%	28.24	2,637	21,044	155
May	483.9	360,000	45.33%	28.09	2,710	25,396	186
Jun	474.6	341,720	45.18%	27.64	2,581	25,520	185
Jul	468.0	348,199	45.15%	27.27	2,631	21,245	152
Aug	465.4	346,281	45.11%	27.15	2,619	20,857	149
Sep	470.3	338,643	45.23%	27.36	2,555	18,516	133
Oct	475.0	353,396	45.36%	27.56	2,658	15,261	111
Nov	482.6	347,505	45.46%	27.94	2,608	13,901	103
Dec	490.8	365,126	45.57%	28.34	2,734	9,909	73
Total	482.0	4,222,139	45.38%	-	31,748	207,255	1,511

CCGT with Inlet Evaporative Cooler System

The evaporative cooler has lesser effectiveness compared to the fogging system. It increases the IPD of the GT due to the additional parts in the inlet duct of the GT. However, the system does not need a demineralized water supply, and it could be operated on a lower water quality, but it requires more water quantity due to the drains. Tables 27 and 28 show the results of the evaporative cooler case.

Table (27): CCGT with Inlet Evaporative Cooler System Results Summary

Month	A. T (°C)	R. H (%)	C.I.T (°C)	C.I RH (%)	Water Consumption Due to Evaporation (m ³)	Total Plant Water Consumption Due to Evaporation and Drain (m ³)
Jan	18.5	54.6	13.7	93.9	5523.8	27,619
Feb	20.4	48.7	14.5	92.8	6012.0	30,060
Mar	23.7	40.8	16.1	91.1	8507.3	42,536
Apr	29.4	29.9	18.7	88.3	11478.8	57,394
May	32.7	26.1	20.3	87.0	13578.2	67,891
Jun	37.0	30.3	24.4	88.0	13200.2	66,001
Jul	37.5	42.1	27.5	91.0	10851.0	54,255
Aug	38.4	43.9	28.6	91.4	10594.8	52,974
Sep	35.6	45.2	26.5	91.7	9533.5	47,667
Oct	31.9	51.6	24.6	93.6	8059.4	40,297
Nov	28.1	49.1	21.0	92.6	7611.6	38,058
Dec	22.0	58.0	17.1	94.4	5539.2	27,696
Total	-	-	-	-	110,490	552,448

**Table (28): CCGT with Inlet Evaporative Cooler System Results
Summary (Continue)**

Month	Total Plant Output (MW)	Total Generated Energy (MWh)	Plant HHV Eff. (%)	Total Plant Gas Flow (kg/s)	Total Inlet Gas Energy (GBTU)	Add. Gen. Energy (MWh)	Added Inlet Gas Energy (GBTU)
Jan	494.6	367,957	45.54	28.58	2,757	7,658	58
Feb	492.9	331,195	45.54	28.48	2,481	8,645	65
Mar	489.6	364,237	45.53	28.30	2,730	12,530	95
Apr	483.2	347,913	45.40	28.01	2,615	17,851	133
May	479.0	356,363	45.30	27.82	2,684	21,758	161
Jun	469.7	338,211	45.16	27.37	2,556	22,011	160
Jul	463.7	345,027	45.12	27.05	2,609	18,073	131
Aug	461.2	343,157	45.08	26.93	2,598	17,732	128
Sep	466.4	335,776	45.20	27.15	2,535	15,650	113
Oct	471.6	350,897	45.34	27.37	2,641	12,762	93
Nov	479.3	345,092	45.43	27.76	2,592	11,488	86
Dec	487.9	363,027	45.54	28.20	2,720	7,810	59
Total	478.2	4,188,852	45.35	-	31,517	173,968	1,280

CCGT with Inlet Mechanical Chiller Cooling System

The mechanical chiller is one of the selected refrigeration systems that work under the vapor compression refrigeration cycle. In the present work, the chiller is driven from the plant generation, causing a drop in plant generation and efficiency. In addition, the added heat exchanger and the mist eliminator to the air inlet duct increase the intake pressure losses, negatively affecting the GT performance. However, the advantage of refrigeration cooling over evaporative cooling is that the wet bulb temperature does not limit the inlet air temperature, and any temperature could be chosen. For the present work, the compressor intake temperature set point has been selected to be 15 °C. Tables 29 and 30 show the results of the mechanical chiller case.

**Table (29): CCGT with Inlet Mechanical Chiller Cooling System
Results Summary**

Month	A. T (°C)	R. H (%)	C.I.T (°C)	C.I RH (%)	Cooling Load Per GT (MW)	Input Power Per GT (MW)
Jan	18.5	54.6	15.0	68.2	1.9	0.5
Feb	20.4	48.7		68.4	2.9	0.8
Mar	23.7	40.8		70.1	4.6	1.3
Apr	29.4	29.9		71.9	7.5	2.1
May	32.7	26.1		75.7	9.2	2.6
Jun	37.0	30.3		100.0	11.3	3.2
Jul	37.5	42.1		100.0	11.8	3.4
Aug	38.4	43.9		100.0	12.3	3.5
Sep	35.6	45.2		100.0	10.9	3.1
Oct	31.9	51.6		100.0	9.0	2.6
Nov	28.1	49.1		100.0	6.9	2.0
Dec	22.0	58.0		90.0	3.7	1.1
Total	-	-	-	-	7.7	2.2

**Table (30): CCGT with Inlet Mechanical Chiller Cooling System
Results Summary (Continue)**

Month	Total Plant Net Output (MW)	Total Gen. Energy (MWh)	Plant HHV Eff. (%)	Total Plant Gas Flow (kg/s)	Total Inlet Gas Energy (GBTU)	Add.Gen .Energy (MWh)	Added Inlet Gas Energy (GBTU)
Jan	489.63	364,285	45.44%	28.36	2735.76	3,986	36.27
Feb	489.07	328,652	45.38%	28.36	2471.01	6,102	54.20
Mar	488.07	363,124	45.28%	28.36	2736.14	11,417	101.10
Apr	485.24	349,375	45.01%	28.37	2648.44	19,312	166.55
May	483.48	359,708	44.83%	28.38	2737.69	25,103	214.17
Jun	481.82	346,907	44.58%	28.44	2655.35	30,707	259.54
Jul	481.34	358,118	44.53%	28.44	2743.86	31,163	265.11
Aug	480.68	357,626	44.47%	28.44	2743.86	32,202	273.98
Sep	482.64	347,498	44.65%	28.44	2655.35	27,372	233.96
Oct	485.03	360,861	44.88%	28.44	2743.86	22,726	196.03
Nov	487.31	350,864	45.09%	28.44	2655.35	17,260	149.75
Dec	489.69	364,327	45.34%	28.42	2741.54	9,109	80.26
Total	485.31	4,251,344	44.96%	-	32,268	236,460	2,031

CCGT with Inlet Absorption Chiller Cooling System

The absorption chiller is another refrigeration cooling system used in this research. The main difference between the mechanical and absorption chillers is that the gas compressor has been replaced by an absorption mechanism that requires heat instead of electricity to be driven.

In this research, two heat sources have been selected. Firstly, an internal source, which is an extracted LP steam from the HRSG. Secondly, a waste heat source. However, the operational drawbacks of absorption chillers are that they have lesser COP than mechanical chillers and the drop of ST output in case an internal steam source has been used. Tables 31 and 32 show the results of a CCGT with an internal steam absorption chiller, while Tables 33 and 34 show the results of a CCGT with a waste heat absorption chiller.

**Table (31): CCGT with Inlet Internal Steam Absorption Chiller
Cooling System Results Summary**

Month	A. T (°C)	R. H (%)	C.I.T (°C)	C.I RH (%)	Cooling Load per GT (MW)	Input heat per GT (MW)	Extracted LP steam per GT (kg/s)
Jan	18.5	54.6	15	68.2	1.9	1.9	0.8
Feb	20.4	48.7		68.4	2.9	2.9	1.2
Mar	23.7	40.8		70.1	4.6	4.6	2.0
Apr	29.4	29.9		71.9	7.5	7.5	3.2
May	32.7	26.1		75.7	9.2	9.2	3.9
Jun	37.0	30.3		100.0	11.3	11.3	4.8
Jul	37.5	42.1		100.0	11.8	11.8	5.0
Aug	38.4	43.9		100.0	12.3	12.3	5.2
Sep	35.6	45.2		100.0	10.9	10.9	4.6
Oct	31.9	51.6		100.0	9.0	9.0	3.8
Nov	28.1	49.1		100.0	6.9	6.9	2.9
Dec	22.0	58.0		90.0	3.7	3.7	1.6
Total	-	-	-	-	7.7	7.7	3.3

**Table (32): CCGT with Inlet Internal Steam Absorption Chiller
Cooling System Results Summary (Continue)**

Month	Total Plant Output (MW)	Total Generated Energy (MWh)	Plant HHV Efficiency (%)	Total Plant Inlet Gas Flow (kg/s)	Total Inlet Gas Energy (GBTU)	Additional Generated Energy (MWh)	Additional Inlet Gas Energy (GBTU)
Jan	489.94	364,512	45.46%	28.36	2735.76	4,213	36.27
Feb	489.53	328,965	45.43%	28.36	2471.01	6,415	54.20
Mar	488.91	363,751	45.36%	28.36	2736.14	12,044	101.10
Apr	486.73	350,444	45.15%	28.37	2648.44	20,382	166.55
May	485.35	361,103	45.01%	28.38	2737.69	26,498	214.17
Jun	484.03	348,504	44.78%	28.44	2655.35	32,305	259.54
Jul	483.88	360,004	44.77%	28.44	2743.86	33,050	265.11
Aug	483.35	359,612	44.72%	28.44	2743.86	34,188	273.98
Sep	484.93	349,147	44.87%	28.44	2655.35	29,021	233.96
Oct	486.85	362,219	45.04%	28.44	2743.86	24,083	196.03
Nov	488.67	351,840	45.21%	28.44	2655.35	18,235	149.75
Dec	490.29	364,777	45.40%	28.42	2741.54	9,560	80.26
Total	486.86	4,264,878	45.10%	-	32,268	249,994	2,031

**Table (33): CCGT with Inlet Waste Heat Absorption Chiller Cooling
System Results Summary**

Month	A. T (°C)	R. H (%)	C.I.T (°C)	C.I RH (%)	Cooling Load Per GT (MW)	Input heat per GT (MW)
Jan	18.5	54.6	15	68.2	1.9	1.9
Feb	20.4	48.7		68.4	2.9	2.9
Mar	23.7	40.8		70.1	4.6	4.6
Apr	29.4	29.9		71.9	7.5	7.5
May	32.7	26.1		75.7	9.2	9.2
Jun	37.0	30.3		100.0	11.3	11.3
Jul	37.5	42.1		100.0	11.8	11.8
Aug	38.4	43.9		100.0	12.3	12.3
Sep	35.6	45.2		100.0	10.9	10.9
Oct	31.9	51.6		100.0	9.0	9.0
Nov	28.1	49.1		100.0	6.9	6.9
Dec	22.0	58.0		90.0	3.7	3.7
Total	-	-	-	-	7.7	7.7

Table (34): CCGT with Inlet Waste Heat Absorption Chiller Cooling System (Continue)

Month	Total Plant Output (MW)	Total Generated Energy (MWh)	Plant HHV Efficiency (%)	Total Plant Gas Flow (kg/s)	Total Inlet Gas Energy (GBTU)	Additional Generated Energy (MWh)	Additional Inlet Gas Energy (GBTU)
Jan	490.71	365,086	45.53%	28.36	2735.76	4,787	36.27
Feb	490.72	329,762	45.54%	28.36	2471.01	7,213	54.20
Mar	490.71	365,088	45.53%	28.36	2736.14	13,382	101.10
Apr	489.54	352,470	45.41%	28.37	2648.44	22,407	166.55
May	488.71	363,601	45.32%	28.38	2737.69	28,997	214.17
Jun	488.28	351,564	45.18%	28.44	2655.35	35,364	259.54
Jul	488.08	363,130	45.16%	28.44	2743.86	36,176	265.11
Aug	487.70	362,851	45.12%	28.44	2743.86	37,427	273.98
Sep	488.84	351,964	45.23%	28.44	2655.35	31,838	233.96
Oct	490.17	364,684	45.35%	28.44	2743.86	26,548	196.03
Nov	491.28	353,718	45.45%	28.44	2655.35	20,114	149.75
Dec	491.83	365,919	45.54%	28.42	2741.54	10,702	80.26
Total	489.71	4,289,837	45.36%	-	32,268	274,953	2,031

4.5 Technical Analysis

This section will provide an overall comparison and discussion of the obtained results by the selected cooling systems and their impact on the CCGT performance.

4.5.1 Power Output

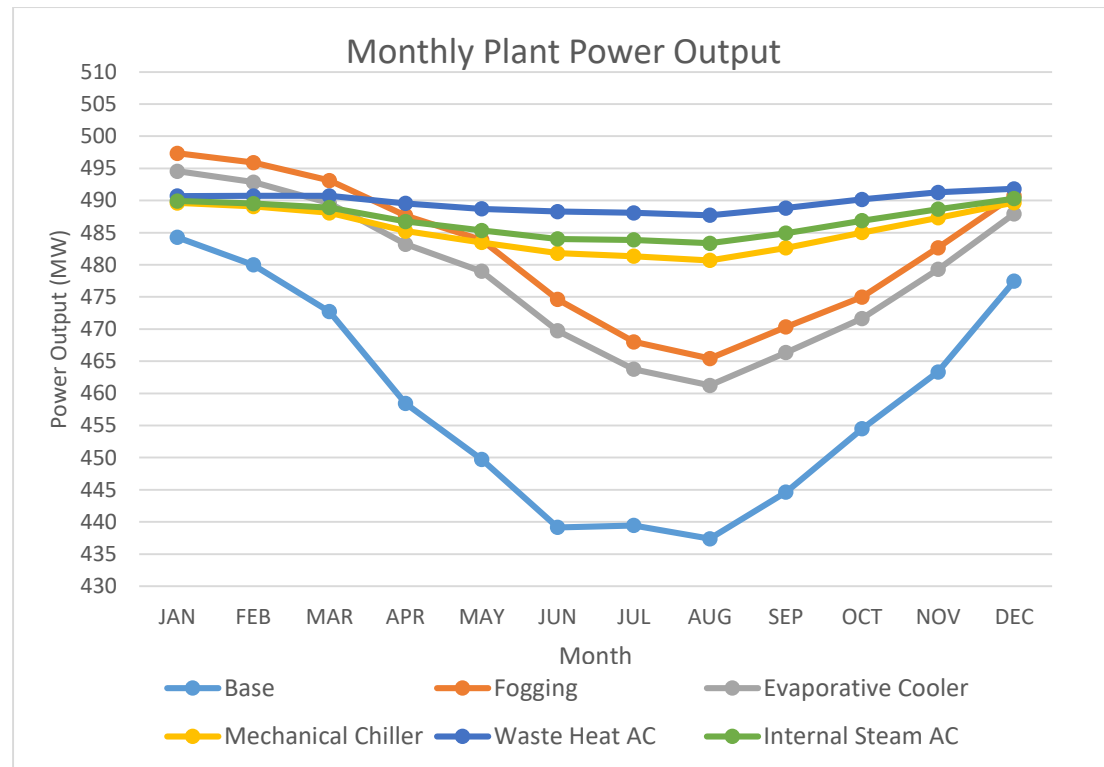


Fig 43: Monthly plant power output as per the used cooling system

Figure 43 shows the monthly plant power output as per the used cooling system. It can be seen that the refrigeration cooling systems have a stable power output throughout the year, which is due to the compressor inlet set point (15 °C). On the other hand, both evaporative cooling systems showed an extreme output drop during summer, which is due to the limitation of the wet bulb temperature. In addition, the evaporative cooling systems generated more output than the refrigeration cooling systems in the first two months of the year because they achieved a lower compressor intake temperature, as shown in the previous section.

Moreover, the fogging system achieved the maximum power output in March among all other systems because it is the only system not causing an additional intake pressure loss, and it achieved a compressor inlet temperature of 15 °C.

The refrigeration cooling systems showed a drop in output during the summer, which is due to the high ST exhaust pressure and the increase in power and heat requirement by the mechanical and internal steam absorption chillers, respectively. Figure 44 shows the condenser pressure of each system, while Tables 30 and 32 show the needed power and heat for the mechanical chiller and LP steam for the internal steam absorption chiller, respectively.

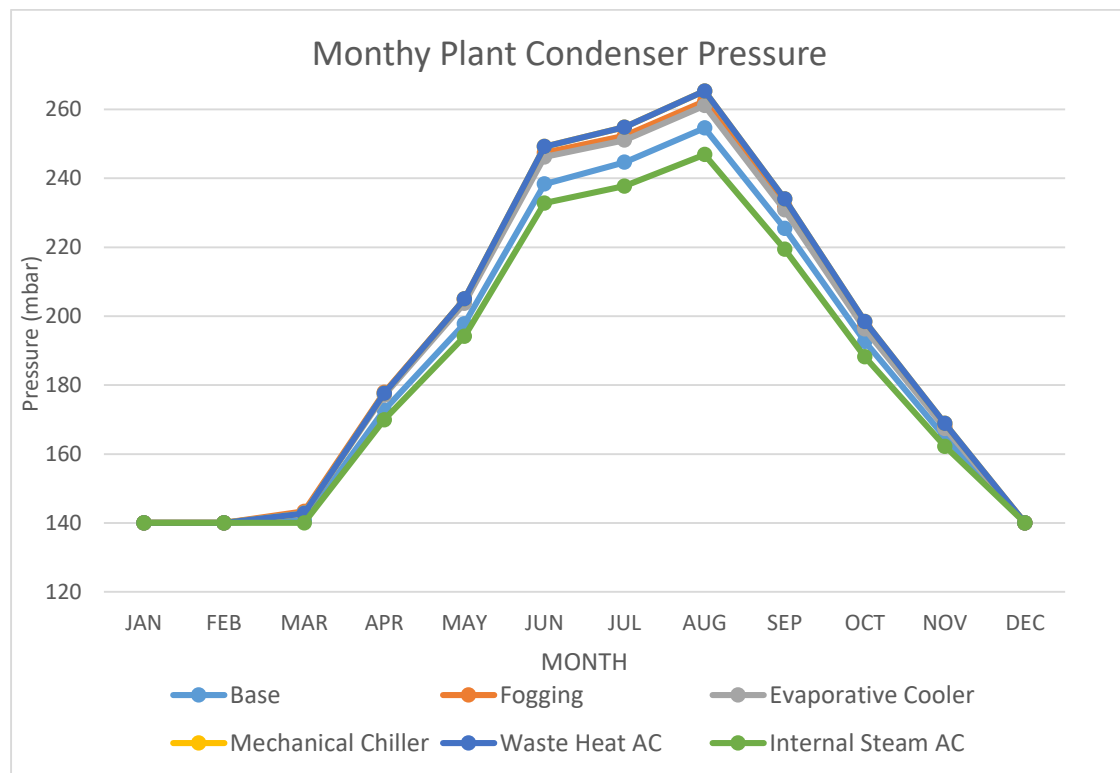


Fig. 44: Monthly plant condenser pressure as per the used cooling system

All cooling systems cause an increase in the condenser pressure, except the internal steam absorption chiller. The reason behind this is the increase of the ST exhaust mass flow rate, while it decreased in the internal steam chiller case, causing a drop in condenser pressure, as shown in Figure 45.

In addition, the mechanical and waste heat absorption chiller cases cause an equal ST exhaust mass flow rate, which, therefore, causes an equal condenser pressure.

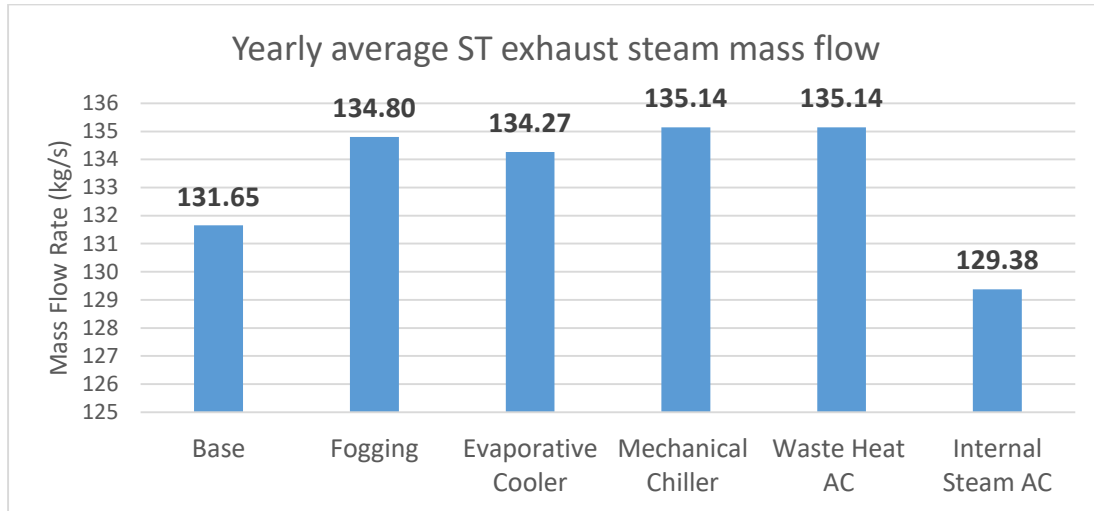


Fig. 45: Yearly average ST exhaust steam mass flow rate per cooling system

By comparing the base case with the cooling systems cases, the following annual additional energy was calculated for each cooling system, as shown in Figure 46. In addition, figure 47 shows the increase in terms of percentage.

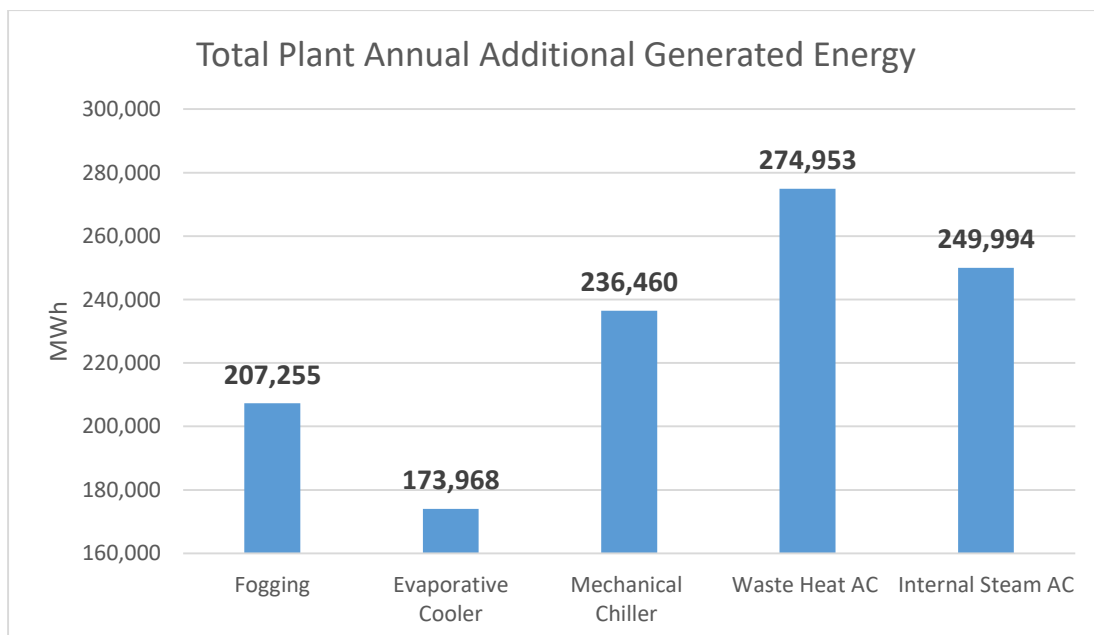


Fig. 46: Total additional plant generated energy per cooling system

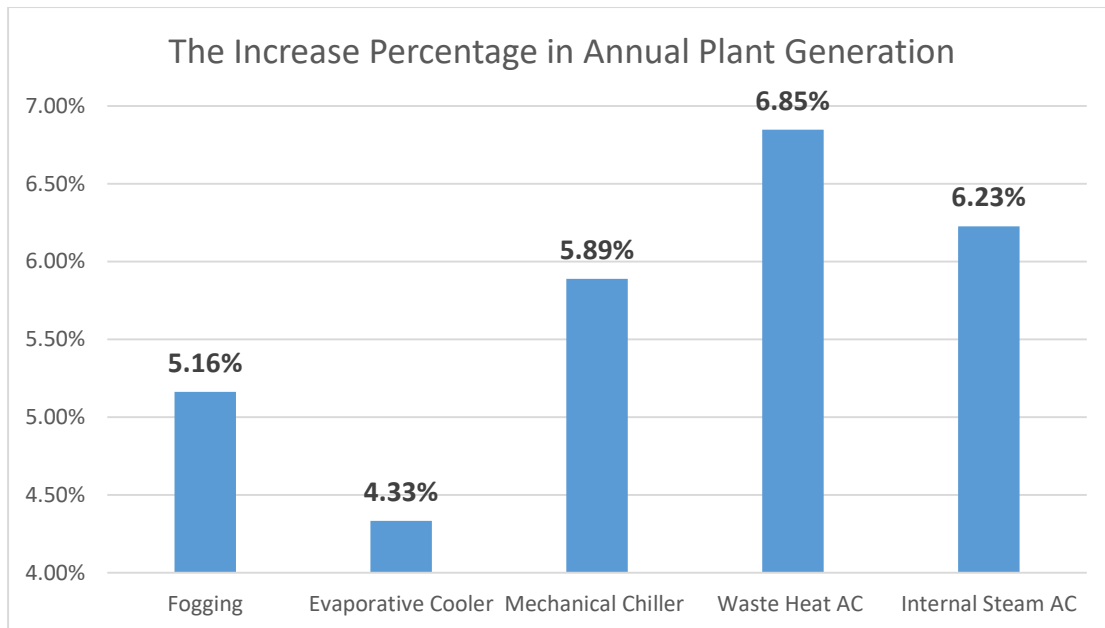


Fig. 47: The increase percentage in annual plant generation per cooling system

Even though the refrigeration cooling systems cool the compressor intake to a fixed value of 15 °C, they generate different amounts of additional energy. This occurs due to the difference in the driving energy of each refrigerating system. However, the results show that the waste heat absorption chiller produced the maximum additional generation because the supplied steam for the chiller is generated from an external source, which does not affect the plant's performance.

The internal steam chiller case has the second maximum additional generation, followed by the mechanical chiller. This concludes that using an internal steam source absorption chiller is more technically feasible than the mechanical chiller, which consumes high electricity.

On the other hand, the evaporated cooling methods generated a lesser amount of additional energy, which is due to the ambient wet bulb temperature limitation.

In addition, the fogging system generated more energy than the evaporative cooler due to the effectiveness and intake pressure loss difference.

4.5.2 Power Plant Efficiency

In addition to the effect of the air intake cooling on the power output of the plant, it also affects its efficiency. Figure 48 shows the monthly plant efficiency that has been obtained by each cooling system.

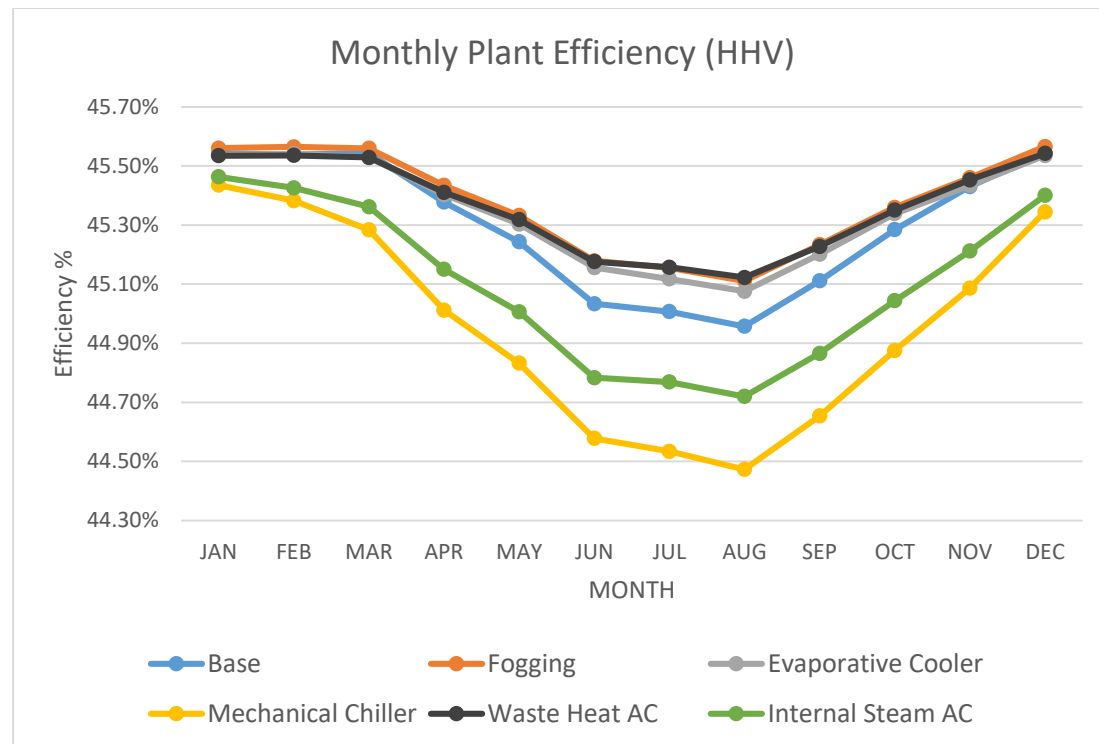


Fig. 48: Monthly plant efficiency per cooling system (HHV)

The results show that the fogging system, evaporative cooler, and waste heat absorption chiller have increased the plant efficiency more than the base case during the summer season. However, the internal steam absorption chiller and the mechanical chiller reduce the plant's efficiency due to their high input energy requirement. Moreover, all cases show a drop in efficiency during the summer caused by the increased condenser pressure, as discussed earlier. Figure 49 shows the annual plant efficiency per cooling system, while Figure 50 shows the change percentage in the annual plant efficiency per cooling system.

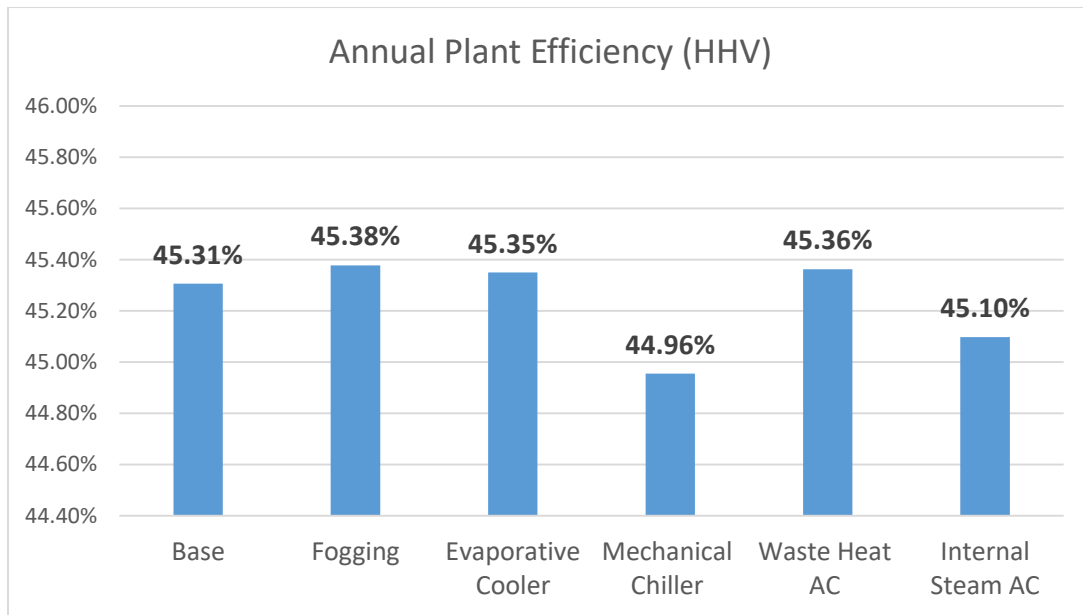


Fig. 49: Annual plant efficiency (HHV)

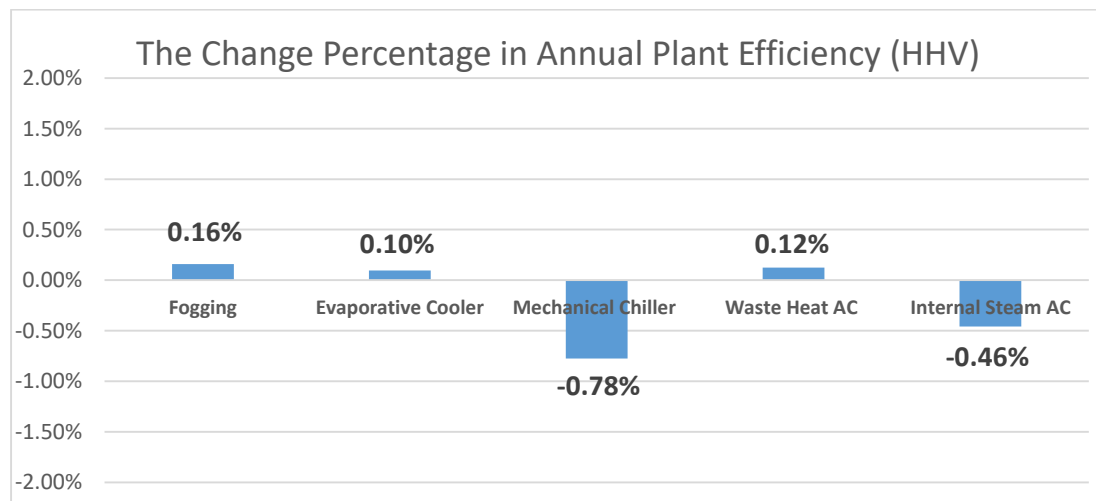


Fig. 50: The change percentage in annual plant efficiency per cooling system (HHV)

The results have shown that the fogging system causes the maximum increase in the annual plant efficiency, followed by the waste heat absorption chiller and evaporative cooler, respectively.

The fogging system does not cause intake pressure losses, and it has lower yearly condenser pressure compared to the waste heat absorption chiller case. Moreover, the evaporative cooler case has the lowest efficiency increase due to the additional intake losses and the lower evaporative cooling effectiveness.

On the other hand, the mechanical chiller reduces the efficiency more than the internal steam absorption chiller because driving the system by the plant's electrical generation is more costly than driving it with the extracted steam.

4.5.3 Additional Gas Consumption

The additional power output is directly proportional to the inlet gas energy to the plant. Figure 51 shows the additional annual inlet gas energy by each cooling system based on the higher heating value of the used gas.

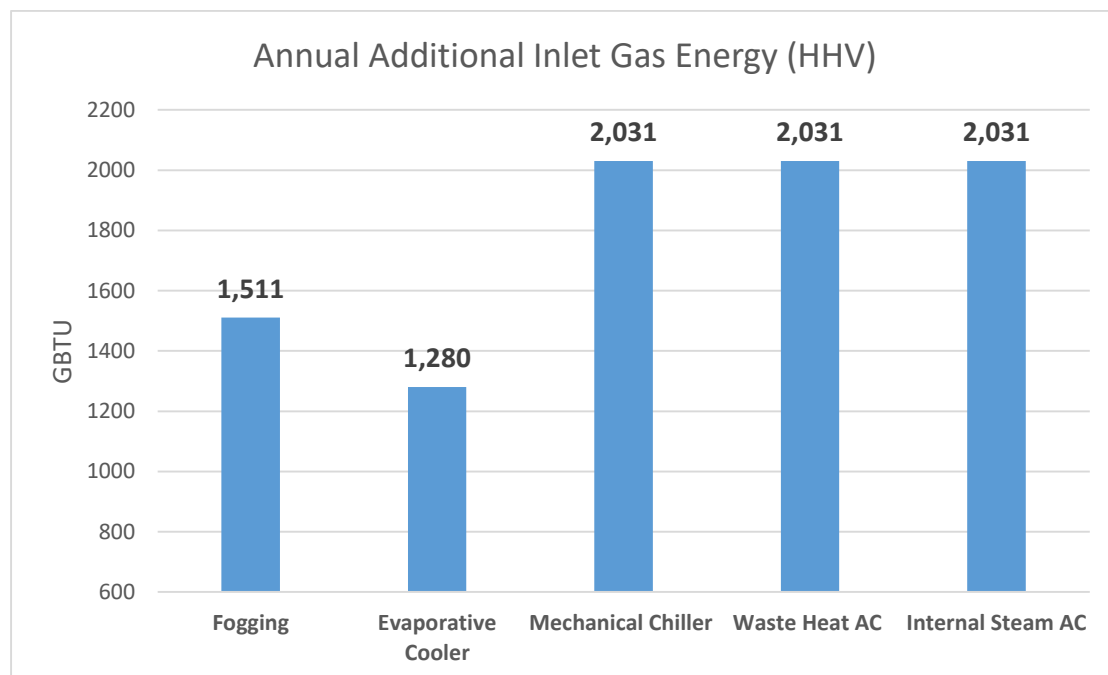


Fig. 51: Additional inlet gas energy per cooling system (HHV)

As seen in Figure 51, all refrigeration cooling systems have the same amount of additional inlet gas energy because they have a fixed compressor air inlet temperature set point, and they cause the same amount of additional inlet pressure losses to the GT.

On the other hand, the additional inlet gas energy by the evaporative cooling systems was less due to the wet bulb temperature limitation, which limits the additional generation.

4.5.4 Technical Analysis Summary

Table 35 summarizes the results of the technical analysis.

Table (35): Technical Analysis summary

Cooling System	Additional Annual Plant Generation (MWh)	Additional Annual Plant Generation (%)	Additional Annual Inlet Gas Energy (GBTU)	Plant HHV Eff. (%)	The Change in Plant Eff. (%)
Fogging	207,255	5.16%	1511	45.38%	0.16 %
Evaporative Cooler	173,968	4.33%	1280	45.35%	0.10%
Mechanical Chiller	236,460	5.89%	2031	44.96%	-0.78%
Waste Heat AC	274,953	6.85%	2031	45.36%	0.12%
Internal Steam AC	249,994	6.23%	2031	45.10%	-0.46%

4.6 Economic Analysis

The technical part of this research has shown the power augmentation capability of each cooling system, where the waste heat absorption chiller generated the maximum additional annual energy, followed by the internal steam absorption chiller, mechanical chiller, fogging system, and evaporative cooler, respectively.

As discussed earlier, a cooling system that generates the highest amount of additional energy is not necessary; it will be the best economical option. Therefore, an overall economic analysis has been done to determine the economic feasibility of each cooling system by using the present worth value, rate of return, and payback period.

Table 36 contains the input values for the economic analysis. The electricity and water tariffs have been collected from Bahrain's electricity

and water authority, and the costs of the cooling systems have been collected from the open literature, as discussed in the previous chapter.

Table (36): Economic Analysis Input Values

Parameter	Value
Interest Rate (%)	6%
N (Years)	20
Electricity Selling Price (\$/kWh)	0.08
Gas Cost (\$/MBTU)	4
Water Cost For Fogging System (\$/ m ³)	3
Water Cost For Evaporative Cooler (\$/ m ³)	2
Fogging System Installation Cost (\$/ kW added)	45
Evaporative Cooler System Installation Cost (\$/ kW added)	50
Mechanical Chiller Installation Cost (\$/ kW added)	250
Absorption Chiller Installation Cost (\$/ kW added)	300
Annual O&M for Fogging System (% of Installation Cost)	3%
Annual O&M for Evaporative Cooler (% of Installation Cost)	4%
Annual O&M for Mechanical Chiller (% of Installation Cost)	6%
Annual O&M for Absorption Chiller (% of Installation Cost)	6%

The simulation results in section 4.4.1 have been utilized to compute the installation cost of each cooling system. Table 37 shows the calculated values of adding two cooling systems for both gas turbines in the plant.

Table (37): Cooling Systems Installation Cost Calculation

Cooling System	Installation Cost (\$/kW Added)	GT Generation at (MW)	Added Power per GT (MW)	Installation Cost (\$) For 2 Units
Base	-	146.071	-	N/A
Fogging	45	172.469	26.40	2,375,820
Evaporative Cooler	50	169.184	23.11	2,311,300
Mechanical Chiller	250	180.733	34.66	17,331,000
Waste Heat A.C	300	180.733	34.66	20,797,200
Internal Steam A.C	300	180.733	34.66	20,797,200

It can be seen from Table 37 that refrigeration cooling systems have a higher initial investment cost compared to evaporative cooling systems, which is one of the drawbacks that have been discussed earlier. Different annual costs are associated with using an air intake cooling system. In the present work, the annual operation and maintenance, additional gas costs, and water costs were considered. Table 38 shows the total annual costs of each cooling system.

Table (38): Cooling Systems Annual Costs

Cooling System	Annual O&M (\$) For 2 Units	Total Plant Annual Additional Gas Cost (\$)	Annual Water Consumption (m ³) For 2 Units	Annual Water Cost (\$) For 2 Units	Total Annual Cost (\$)
Fogging	71,275	6,042,066	124,843	374,529	6,487,869
Evaporative Cooler	92,452	5,120,042	552,448	1,104,897	6,317,390
Mechanical Chiller	1,039,860	8,123,722	-	-	9,163,582
Waste Heat A.C	1,247,832	8,123,722	-	-	9,371,554
Internal Steam A.C	1,247,832	8,123,722	-	-	9,371,554

It can be seen from Table 38 that the majority of the annual cost is due to the additional gas cost for all cases. Moreover, the absorption chiller also has the highest annual cost in addition to its highest installation cost.

Even though the absorption chillers have the highest initial and annual costs, these costs could be recovered by their additional generated energy. As per the technical analysis, both internal and external absorption chillers have the two highest additional generation, which means the highest amount of revenues. Table 39 shows the annual revenues and the net cash flow of each cooling system.

Table (39): Cooling Systems Annual Revenues and Net Cash Flow

Cooling System	Annual Revenue (\$)	Annual Net Cash Flow (\$)
Fogging	16,580,394	10,092,525
Evaporative Cooler	13,917,456	7,600,066
Mechanical Chiller	18,916,803	9,753,221
Internal Steam A.C	19,999,521	10,627,967
Waste Heat A.C	21,996,236	12,624,682

As discussed, waste heat and internal steam absorption chillers have the two highest annual revenues and, therefore, the highest net cash flow. The fogging system has the third highest net cash flow, which is due to its low annual cost and the high additional generation energy.

To have a better interpretation of the economic evaluation, three economic metrics have been used: net present value, rate of return, and payback period. Microsoft Excel has been used to solve the economic model, using its built-in functions. Figure 52 shows the initial investment costs, the net

cash flows, and, finally, the results of the economic analysis for all presented cooling systems.

	A	B	C	D	E	F	G
20							
21		year	Fogging System	Evaporative Cooler	Mechanical Chiller	Internal Steam AC	Waste Heat AC
22		0	\$ (2,375,820.00)	\$(2,311,300.00)	\$ (17,331,000.00)	\$ (20,797,200.00)	\$(20,797,200.00)
23		1	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
24		2	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
25		3	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
26		4	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
27		5	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
28		6	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
29		7	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
30		8	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
31		9	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
32		10	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
33		11	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
34		12	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
35		13	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
36		14	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
37		15	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
38		16	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
39		17	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
40		18	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
41		19	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
42		20	\$ 10,092,524.97	\$ 7,600,065.54	\$ 9,753,220.62	\$ 10,627,966.68	\$ 12,624,682.20
43		NPV	\$113,384,646	\$84,860,853	\$94,537,672	\$101,104,741	\$124,006,910
44		IRR	425%	329%	56%	51%	61%
45		P.B	0.24	0.30	1.78	1.96	1.65

Fig. 52: Economic analysis results in Microsoft Excel

4.6.1 Net Present Value

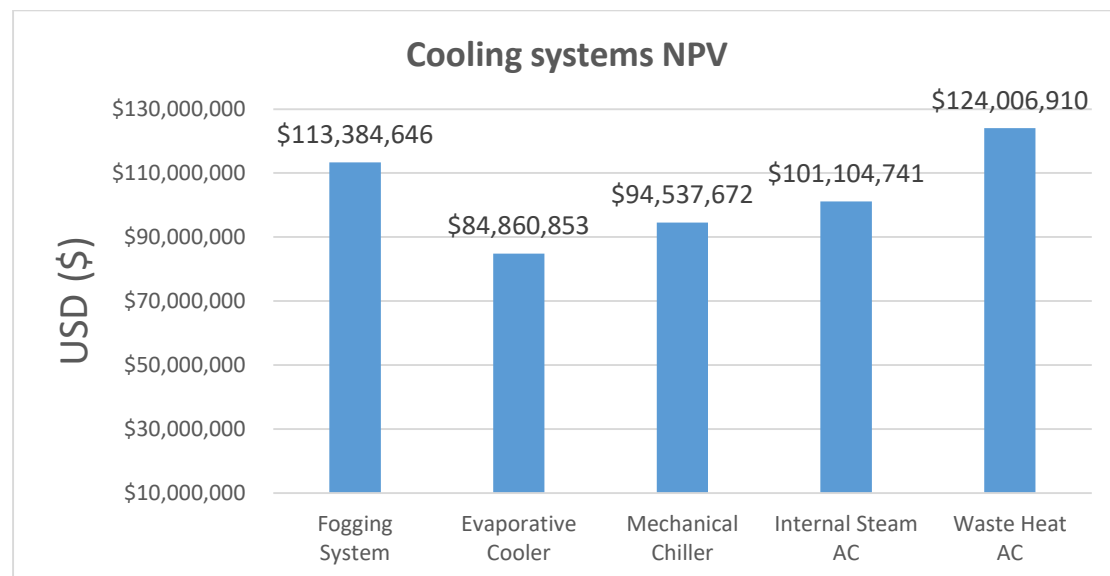


Fig. 53: Cooling systems net present values

Figure 53 compares the NPV of the presented cooling systems. The figure shows that the waste heat absorption chiller has the largest NPV, followed by the fogging system, internal steam absorption chiller, mechanical chiller, and evaporative cooler, respectively.

Even though the waste heat absorption chiller has the largest installation and annual cost, it has the highest NPV. This is because it uses a waste heat source to reduce the compressor intake temperature to 15 °C, which maximizes its net cash flow, as seen in Table 39.

The internal steam absorption and mechanical chillers also reduce the compressor intake temperature to 15 °C, but by using internal energy sources, which causes a reduction in their net cash flows.

On the other hand, the fogging system has the second largest NPV due to its low installation cost and high net cash flow.

Although the evaporative cooler has a low installation cost, its NPV is the lowest because it has the lowest net cash flow.

4.6.2 IRR and PB period

The IRR and the PB period have been used also as other economic tools. Figures 54 and 55 show the IRR and PB of the presented cooling systems.

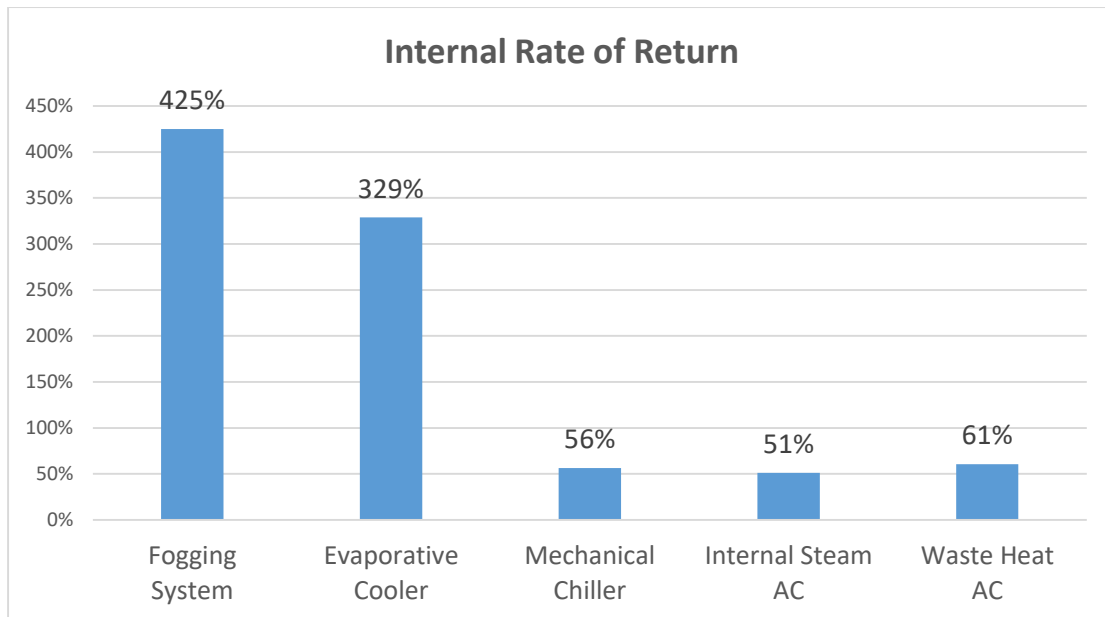


Fig. 54: Cooling systems Internal Rate of Return

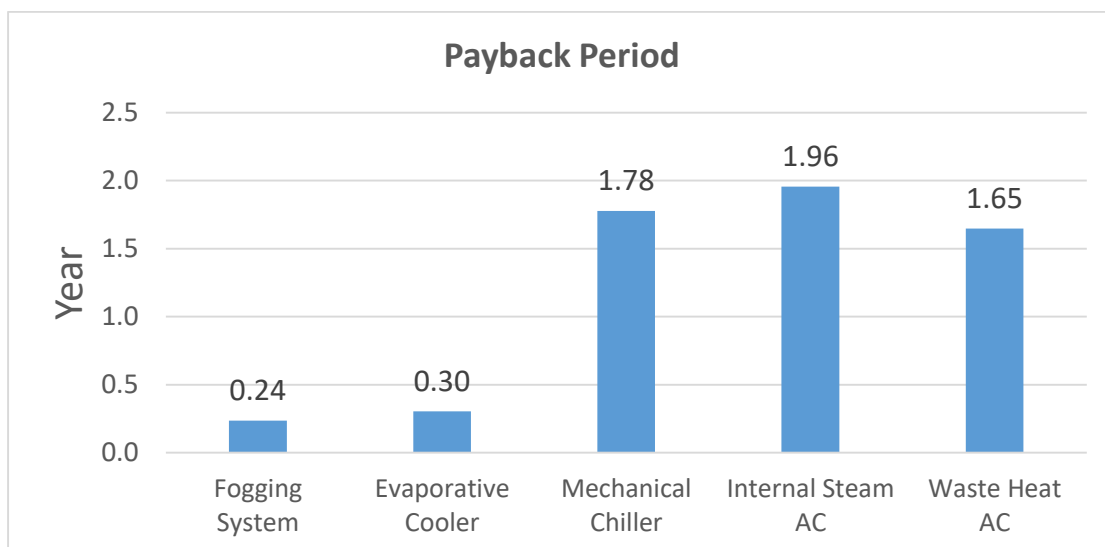


Fig. 55: Cooling systems payback periods

The results show that the fogging system and the evaporative cooler have the highest IRR and the lowest PB, among others. Therefore, the investment in evaporative cooling technologies suits those requiring a faster return on investment. However, due to the higher installation cost of the refrigeration cooling systems, the results show that they have a lower IRR and a higher PB than the evaporative cooling systems.

4.7 The Selection

The selection process among the presented cooling systems is mutually exclusive, where only one alternative can be selected.

Different factors need to be considered during the selection, such as available budget, additional gas flow restrictions, water availability, and the owner's objectives.

In general, it has been recommended to depend on the NPV to select between investments because it can provide a better profitability measure and has more realistic assumptions (Vaidya, 2018). Therefore, for the present work, the selection process is based mainly on the NPV of each cooling system and assuming that there are no constraints on the available budget and the time at which the investment will be recovered. Table 40 below summarizes the results after the economic analysis.

Table (40): Economic Analysis Summary

Cooling System	NPV	IRR	PB
Fogging System	\$113,384,646	425%	0.24
Evaporative Cooler	\$84,860,853	329%	0.30
Mechanical Chiller	\$94,537,672	56%	1.78
Internal Steam A.C	\$101,104,741	51%	1.96
External Steam A.C	\$124,006,910	61%	1.65

As per the results, the waste heat absorption chiller has the largest NPV among all others, which makes it an attractive selection. However, it requires an enormous waste heat amount.

Operating the absorption chiller on an internal steam source reduces its NPV by \$22,902,170, which makes it an unattractive alternative considering other presented cooling alternatives.

The evaporative cooler and the mechanical chiller have the two lowest net present values, making them also unattractive investments. On the other hand, the fogging system has the second largest NPV, shortest PB, and highest IRR, which makes it the best option only in case there is no feasible waste heat source to drive the absorption chiller in the plant.

Chapter Five

Conclusion and Recommendation

5.1 Conclusion

5.2 Recommendation

Chapter 5: Conclusion & Recommendation

5.1 Conclusion

In the present research, the technical and economic feasibility of using different GT inlet air cooling systems for CCGT was investigated based on Bahrain operating conditions. The selected plant for this research consists of two gas turbines, two heat recovery steam generators, and one steam turbine. The plant has been modeled in Ebsilon Professional software using the original equipment manufacturer data, and it has been validated at two different ambient conditions.

The effect of the ambient temperature on CCGT performance was investigated by ranging the ambient temperature. The results have shown that by increasing the ambient temperature from 15 °C to 45 °C, the total plant power output and efficiency will be reduced by 14 % and 2.2%, respectively. This significant impact was mainly due to the reduction of GT inlet air density, the increase in the air compressor's work, and the increase of steam turbine exhaust pressure due to the air-cooled condenser limitation.

To reduce the impact of ambient temperature on the plant, evaporative and refrigeration cooling methods were proposed to cool the intake of the GT. The evaporative cooling method has been represented by using a fogging system and an evaporative cooler, while the refrigeration cooling method has been represented by using a mechanical chiller and absorption chiller. Moreover, it has been assumed that the electricity requirement to drive the mechanical chiller is supplied internally from the plant, while the heat requirement for the absorption chiller was assumed to be supplied from two sources: an internal low-pressure steam extracted from the HRSG and a waste heat source. Therefore, six cases have been created: CCGT without

cooling (Base case), CCGT with fogging systems, CCGT with evaporative coolers, CCGT with mechanical chillers, CCGT with internal steam absorption chillers, and CCGT with waste heat absorption chiller.

Bahrain's climate data for the year 2022 has been used as input variables to simulate the annual behavior of the six mentioned cases. In addition, the amount of the annual additional generation and gas consumption were calculated by comparing the base case with each remaining case.

The results have shown that the waste heat absorption chiller, internal steam absorption chiller, mechanical chiller, fogging system, and evaporative cooler increase the annual plant generation by 6.85%, 6.23%, 5.89%, 5.16%, and 4.33% respectively. Moreover, it has been noticed that the fogging system, waste heat absorption chiller, and evaporative cooler improve the annual plant efficiency by 0.16%, 0.12%, and 0.10% respectively. However, the mechanical and the internal steam absorption chillers reduce the annual plant efficiency by 0.78% and 0.46%, respectively, due to the high input energy supplied internally from the plant.

The technical analysis has been followed by an economic analysis to investigate the profitability of each cooling system, which was based on an equal life cycle of 20 years and an interest rate of 6%.

The results have shown that the waste heat absorption chiller case has the highest NPV of \$124 M with an IRR and PB of 61% and 1.65 Years, respectively. In addition, the fogging system case has the second-highest NPV of \$113.4 M with an IRR and PB of 425% and 0.24 years respectively. However, mechanical chillers, internal steam absorption chillers, and evaporative coolers were economically unattractive compared to the waste heat absorption chiller and fogging system cases.

To sum up, the presented research has concluded that using waste heat absorption chillers to cool the air intake of gas turbines in combined cycle power plants can generate the most significant profit in Bahrain, subject to the availability of a sufficient waste heat source. However, in case of unavailability of a waste heat source in the plant, the fogging system should be prioritized, followed by an internal steam absorption chiller, mechanical chiller, and evaporative cooler, respectively.

5.2 Recommendation

The presented research has shown that ambient temperature is a critical parameter that impacts CCGT plants. However, the following recommendations need to be taken into consideration:

- Considering Bahrain's yearly climate profile, using a GT inlet air cooling system is mandatory to reduce the effect of the high ambient temperature on combined cycle gas turbine plants.
- The power plant owners in Bahrain should invest in adding absorption chillers to cool the air intake of gas turbines only if a sufficient waste heat source is available in the plant. However, the investment should be made in adding fogging systems if there are no feasible waste heat sources in the plant.
- Future work should investigate the practical feasibility of utilizing the waste heat in HRSG exhaust gas and steam turbine condensers for driving absorption chillers in CCGT plants.

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Appendices

7.1 Models Calculation Flow Charts

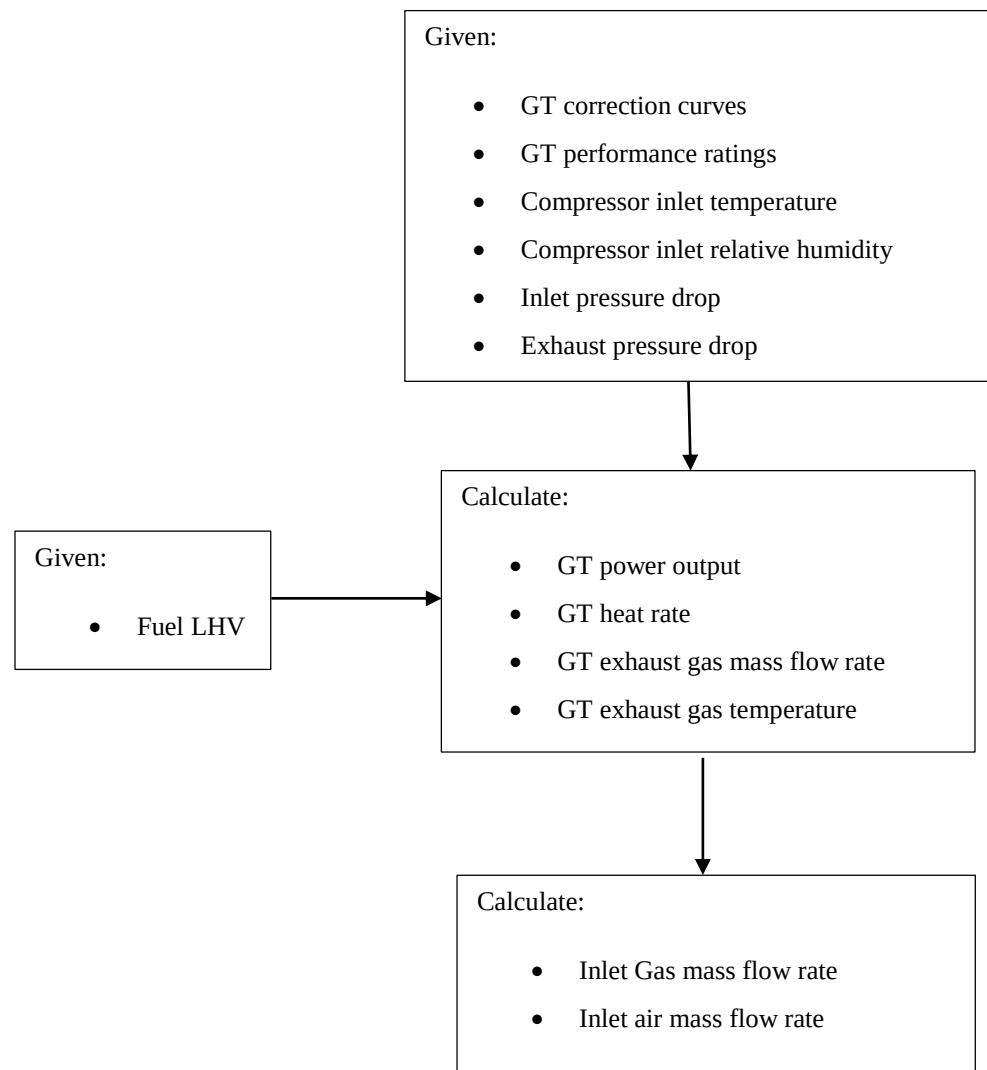
7.2 Simulation Results in Epsilon

APPENDICES

7.1 Appendix A: Models Calculation Flow Chart

This appendix shows the calculation flow charts of the models used in the presented work.

7.1.1 Gas Turbine Model



7.1.2 HRSG Model

Given:

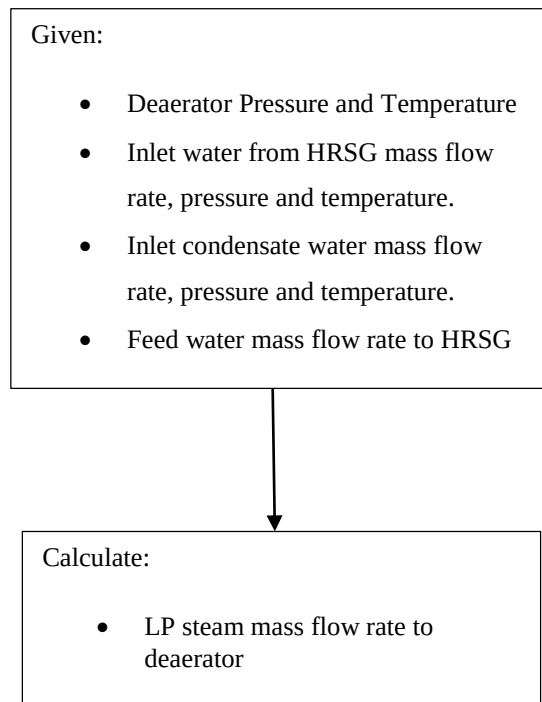
- HRSG inlet gas temperature
- HRSG inlet gas mass flow rate
- HP inlet feed water pressure
- HP inlet feed water temperature
- HP drum pressure
- LP inlet feed water pressure
- LP inlet feed water temperature
- LP drum pressure
- Extracted LP water mass flow rate to the deaerator
- Heat exchangers surface areas
- Heat transfer coefficients



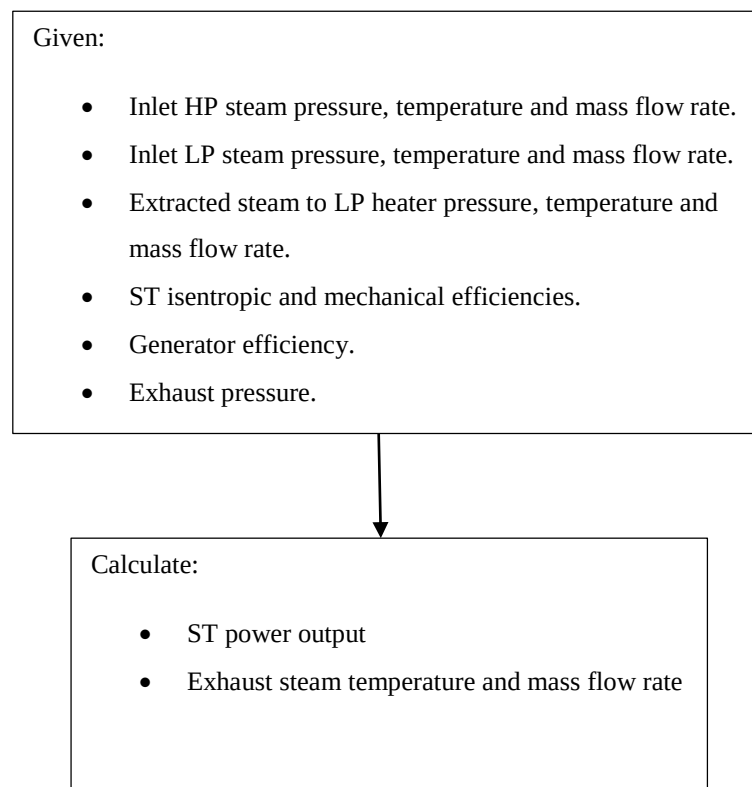
Calculate:

- HP steam outlet mass flow rate
- HP steam outlet temperature
- LP steam outlet mass flow rate
- LP steam outlet temperature
- Exhaust gas Temperature

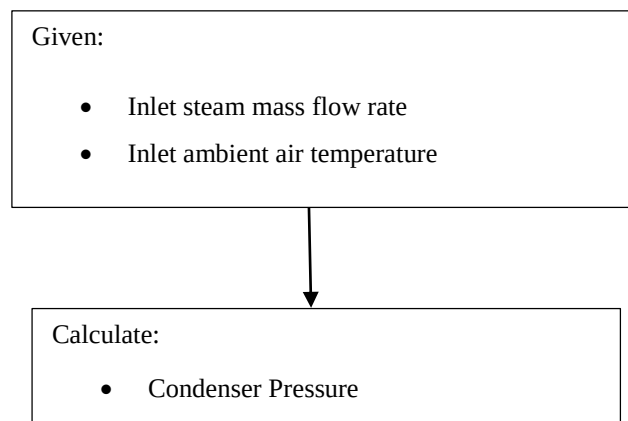
7.1.3 Deaerator Model



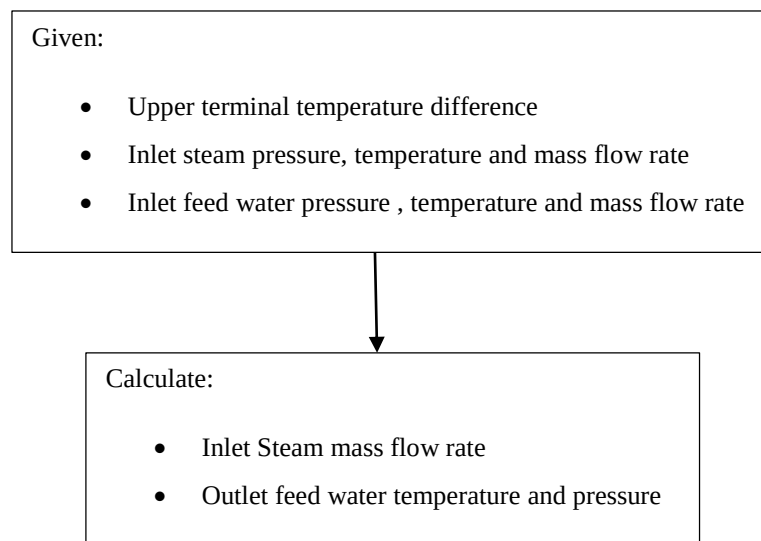
7.1.4 Steam turbine Model



7.1.5 Air cooled condenser Model



7.1.6 Feed water pre heater Model



7.1.7 CCGT plant without cooling (Base Case) model

Given:

- Compressor intake temperature.
- Compressor intake relative humidity.
- Inlet ambient air temperature to condenser.

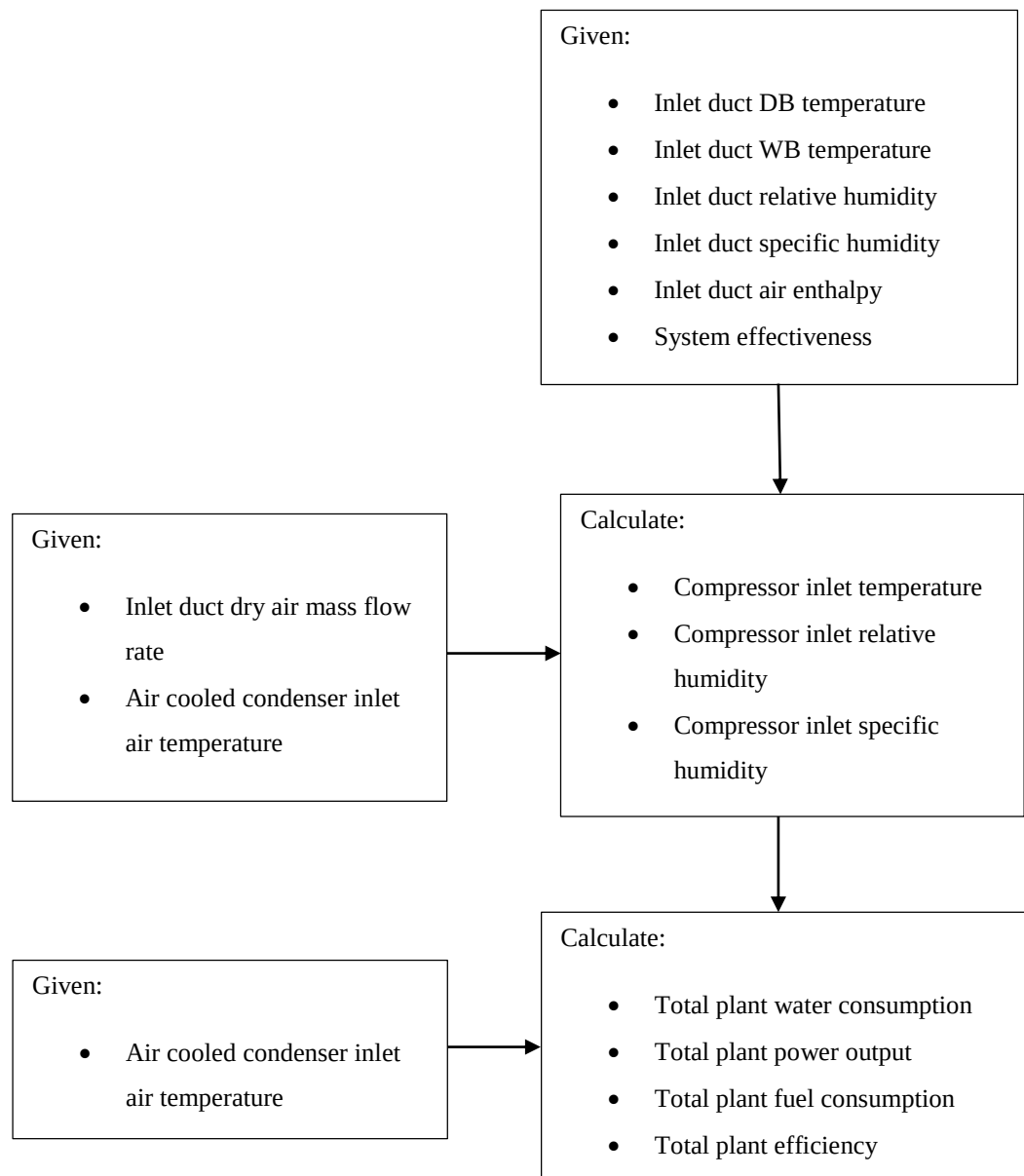


Calculate:

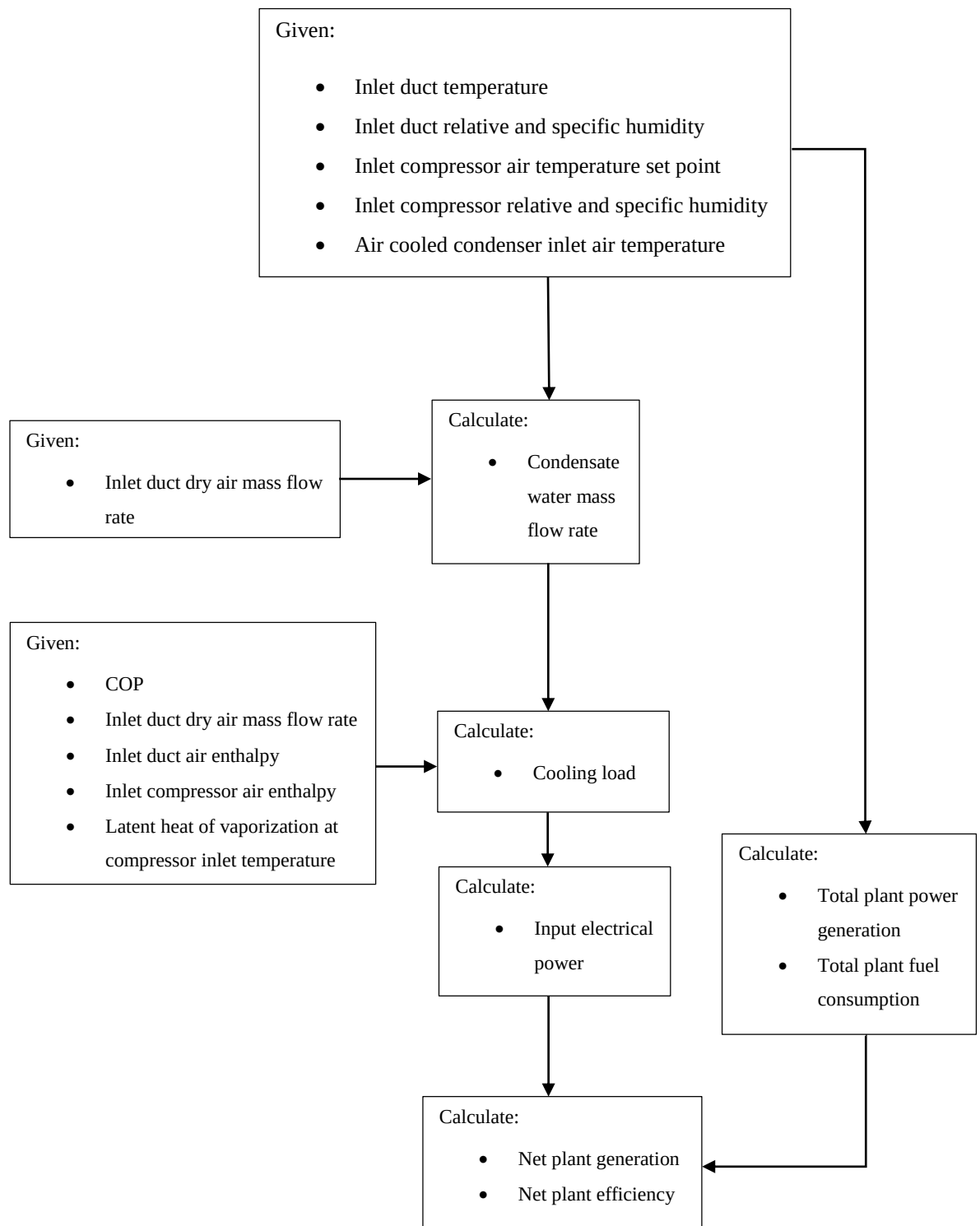
- Total plant power generation
- Total plant fuel consumption
- Total plant efficiency

7.1.8 CCGT plant with evaporative cooling systems model

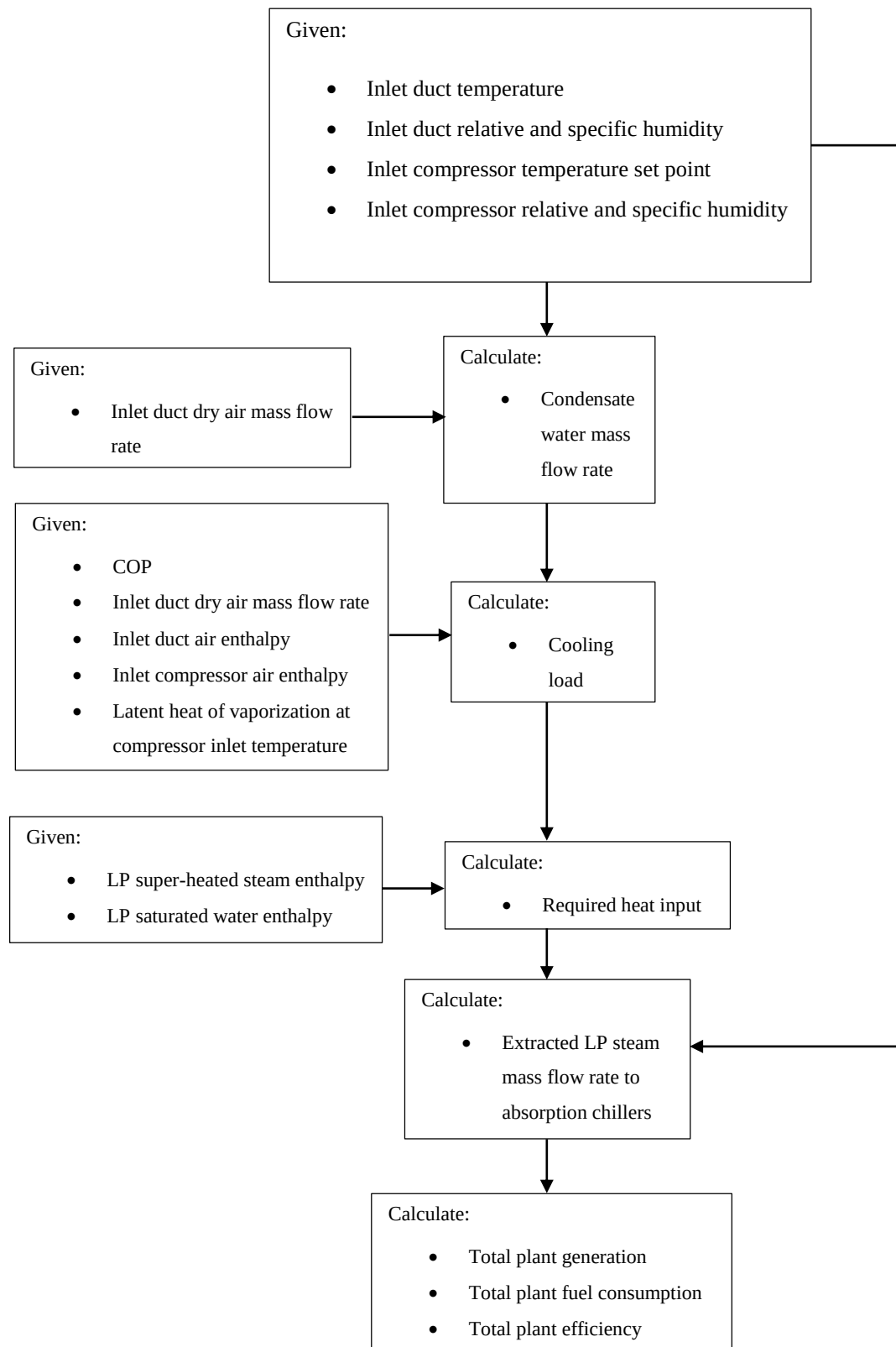
The following calculation flow chart applies to the CCGT plant with a fogging cooling system and the CCGT plant with an evaporative cooler system.



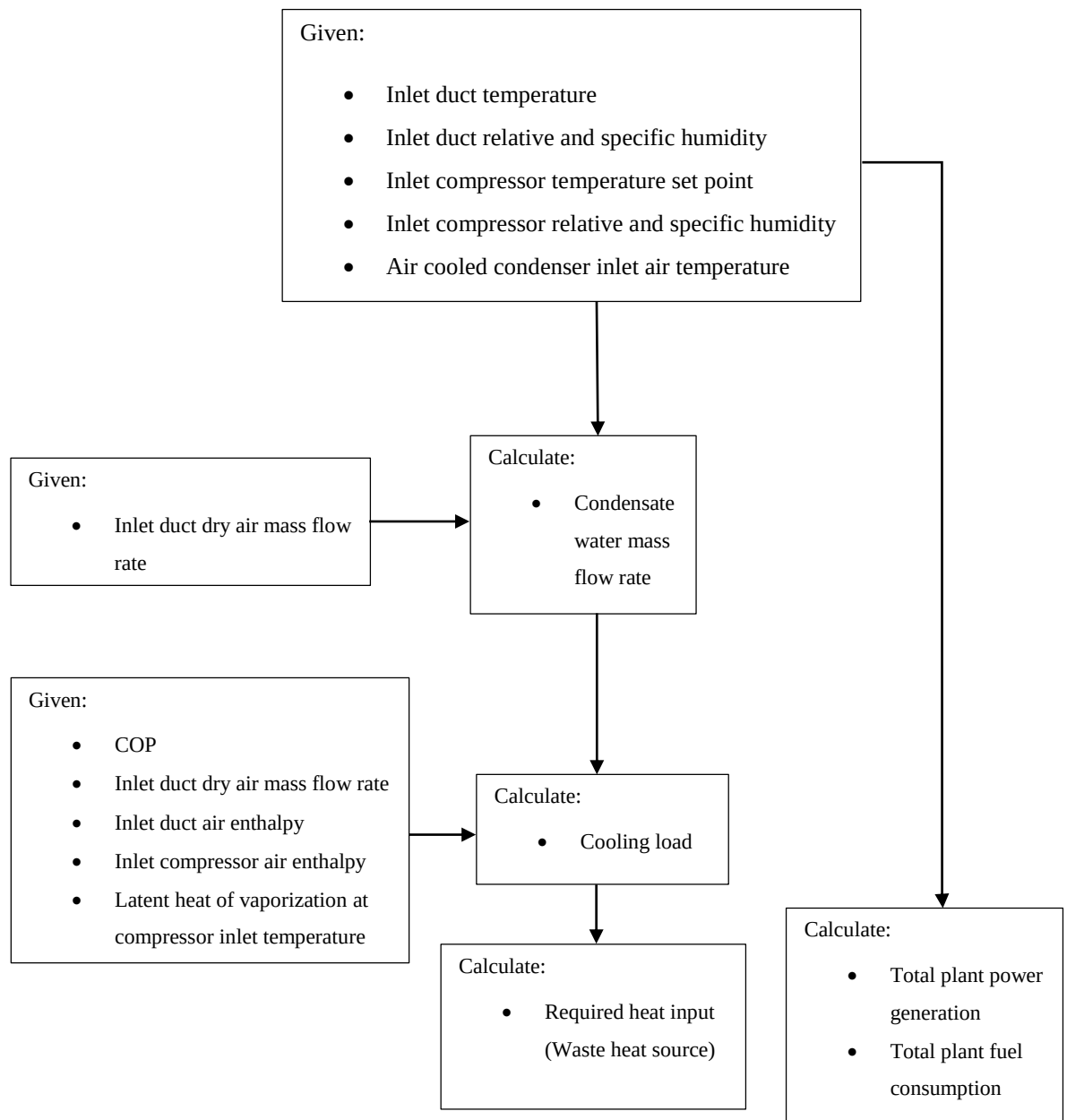
7.1.9 CCGT plant with mechanical chiller



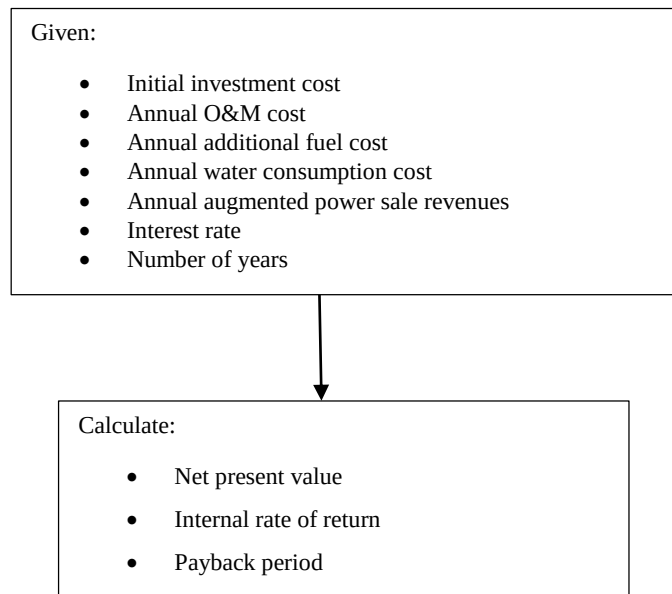
7.1.10 CCPP with inlet internal steam absorption chiller:



7.1.11 CCGT plant with waste heat absorption chiller



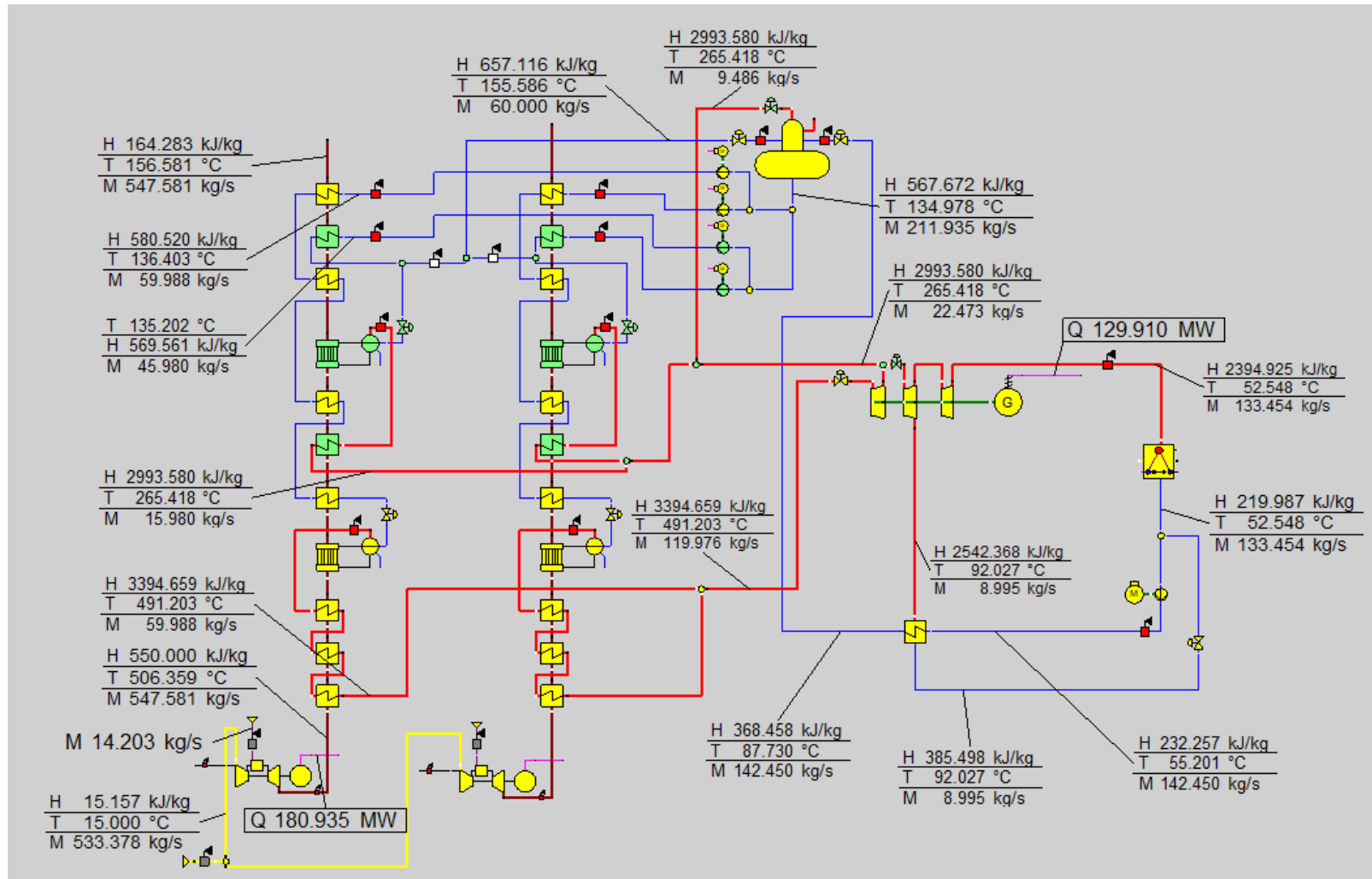
7.1.12 Economic model calculation flow chart for each cooling system:



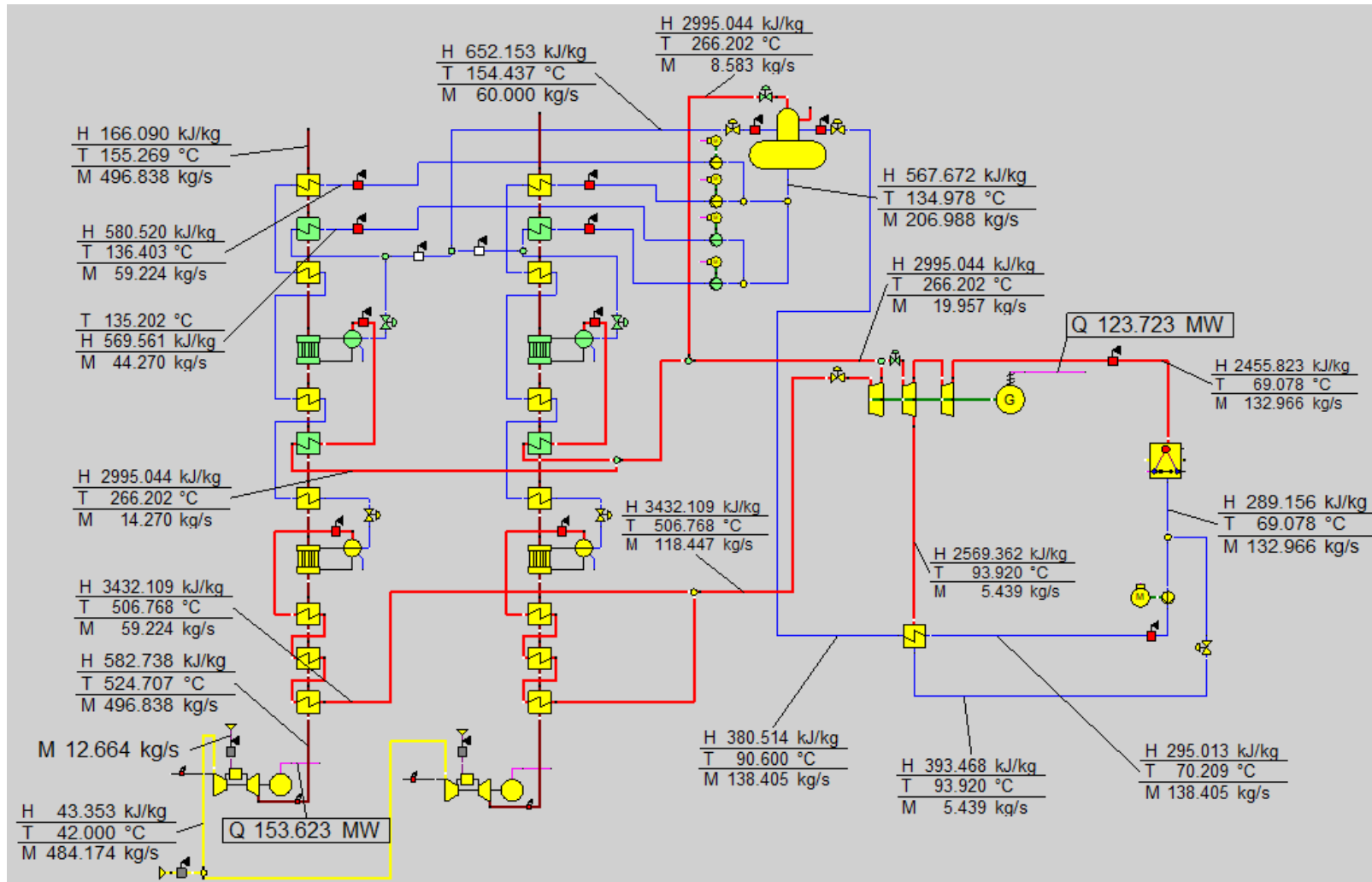
7.2 Appendix B: Simulations Results in Ebsilon

This appendix shows the simulation results of the CCGT plant model in Ebsilon at ambient temperatures of 15 °C and 42 °C, respectively.

7.2.1 CCGT Model Simulation Results at 15 °C



7.2.2 CCGT Model Simulation Results at 42 °C



الملخص

واحدة من العوائق الرئيسية لمحطات التوربينات الغازية ذات الدورة المركبة، هي الحساسية لدرجة الحرارة المحيطة.

تؤدي الزيادة في درجة الحرارة المحيطة إلى تقليل كفاءة المحطة وقدرتها الإنتاجية. في البحرين ، تصل درجة الحرارة المحيطة إلى حوالي 48 درجة مئوية خلال فصل الصيف ، مما يؤثر على أداء محطات الطاقة بشكل سلبي.

استنادًا إلى التحليل الفني والاقتصادي ، بحثت هذه الأطروحة في جدوى إضافة أنظمة تبريد مختلفة لتبريد الهواء الداخل للتوربينات الغازية ذات الدورة المركبة في البحرين، حيث تم اختيار نظام الضباب والمبرد التبخيري لتمثيل طريقة التبريد عن طريق التبخير ، و تم اختيار المبرد الميكانيكي والمبرد الامتصاصي عن طريق البخار الداخلي ، و المبرد الامتصاصي عن طريق الحرارة المهدرة لتمثيل طريقة التبريد بالتبادل الحراري.

تتكون محطة التوربينات الغازية ذات الدورة المركبة المختارة لهذا العمل من توربينين غازيين وتوربين بخاري واحد ، مصممة بطاقة إجمالية تبلغ 491 ميجاوات. تم تصميم المصنع ومحاكاته في برنامج إيسيلون بروفيشنال بناءً على بيانات الشركة المصنعة و بيانات المناخ في البحرين لعام 2022.

أظهر التحليل الفني أن المبرد الامتصاصي عن طريق الحرارة المهدرة ، والمبرد الامتصاصي عن طريق البخار الداخلي ، والمبرد الميكانيكي ، ونظام الضباب ، والمبرد التبخيري زاد من إنتاجية المصنع السنوية للطاقة بمقدار 6.85%، 6.23%، 5.89%، 5.16%، و 4.33% على التوالي. إضافة إلى ذلك ، أدى نظام الضباب والمبرد الامتصاصي عن طريق الحرارة المهدرة والمبرد التبخيري لزيادة كفاءة المصنع السنوية بنسبة 0.16%، 0.12%، و 0.10% على التوالي. من ناحية أخرى ، قلل كل من المبرد الميكانيكي و المبرد الامتصاصي عن طريق البخار الداخلي من كفاءة المصنع السنوية بنسبة 0.78% و 0.46% على التوالي.

أظهرت نتائج التحليل الاقتصادي، أن القيم الحالية الصافية للمبرد الامتصاصي عن طريق الحرارة المهدرة ، ونظام الضباب ، و المبرد الامتصاصي عن طريق البخار الداخلي ، والمبرد الميكانيكي ، والمبرد التبخيري كانت 124 مليون دولار ، و 113 مليون دولار ، و 101 مليون دولار ، و 95 مليون دولار ، و 85 مليون دولار على التوالي.

توصل هذا البحث إلى أن المبرد الامتصاصي عن طريق الحرارة المهدرة هو أفضل بديل لتبريد الهواء الداخل للتوربينات الغازية ذات الدورة المركبة في البحرين بشرط أن يتوفر هناك مصدر حرارة مهتر بشكل كاف. خلاف ذلك ، يجب إعطاء الأولوية لنظام الضباب ، متبوعًا بالمبرد الامتصاصي عن طريق البخار الداخلي ، والمبرد الميكانيكي ، والمبرد التبخيري على التوالي

جامعة البحرين



قسم الهندسة الميكانيكية

كلية الهندسة

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للتوربينات الغازية ذات الدورة المركبة لظروف التشغيل في
مملكة البحرين**

أطروحة مقدمة كجزء من متطلبات الحصول على درجة الماجستير في الإدارة الهندسية

إعداد

الاسم: علي ممدوح محمد الحايكي
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جامعة البحرين

مملكة البحرين

نوفمبر 2023م