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DEVELOPMENT OF A PHOTOVOLTAIC REVERSE OSMOSIS DEMINERALIZATION FOGGING FOR IMPROVED GAS TURBINE GENERATION OUTPUT

by

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Declaration

I, Tariq M. H. Lameen, declare that the content of this thesis represents my own unaided work, and that the thesis has not previously been submitted for academic examination towards any qualification. Furthermore, it represents my own opinions and not necessarily those of the Cape Peninsula University of Technology.

Signed **Date**

Abstract

Gas turbines have achieved widespread popularity in industrial fields. This is due to the high power, reliability, high efficiency, and its use of cheap gas as fuel. However, a major drawback of gas turbines is due to the strong function of ambient air temperature with its output power. With every degree rise in temperature, the power output drops between 0.54 and 0.9 percent. This loss in power poses a significant problem for utilities, power suppliers, and co-generations, especially during the hot seasons when electric power demand and ambient temperatures are high.

One way to overcome this drop in output power is to cool the inlet air temperature. There are many different commercially available means to provide turbine inlet cooling. This dissertation reviews the various technologies of inlet air cooling with a comprehensive overview of the state-of-the-art of inlet fogging systems.

In this technique, water vapour is being used for the cooling purposes. Therefore, the water quality requirements have been considered in this thesis. The fog water is generally demineralized through a process of Reverse Osmosis (RO). The drawback of fogging is that it requires large amounts of demineralized water. The challenge confronting operators using the fogging system in remote locations is the water scarcity or poor water quality availability. However, in isolated hot areas with high levels of radiation making use of solar PV energy to supply inlet cooling system power requirements is a sustainable approach.

The proposed work herein is on the development of a photovoltaic (PV) application for driving the fogging system. The design considered for improved generation of Acaica power plant in Cape Town, South Africa. In addition, this work intends to provide technical information and requirements of the fogging system design to achieve additional power output gains for the selected power plant.

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Nomenclature

List of Abbreviations

AC	Alternative Current
AMCO	American Moistening Company
CC	Combustion Chamber
CCGT	Combined Cycle Gas Turbine
CPDP	Cogeneration Power Desalination Plant
CT	Combustion Turbine
DB	Dry Bulb
DC	Direct Current
DCS	Digital Control System
EPRI	Electric Power Research Institute
GT	Gas Turbine
GTCC	Gas Turbine Combined Cycle
HFO	Heavy Fuel Oil
IPPs	Independent Power Producers
ISO	International Organization for Standardization
LPM	Liter Per Minute

MW	Mega Watt
NG	Natural Gas
PLC	Programmable Logic Controller
PV	Photovoltaic
PVRO	Photovoltaic Reverse Osmosis
RE	Renewable Energy
RH	Relative Humidity
RO	Reverse Osmosis
SA	South Africa
SAPP	South African Pool Power
TDS	Total Dissolved Solids
TIC	Turbine Inlet Cooling
TICA	Turbine Inlet Cooling Association
TIT	Turbine Inlet Temperature
VSD	Variable Speed Driver
WB	Wet Bulb
WBD	Wet Bulb Depression
WBT	Wet Bulb Temperature

Symbols		Units
η	Efficiency	[-]
h	Specific enthalpy	[kJ/kg]
HR	Heat rate	[kJ/kWh]
P	Pressure	[bar]
$\dot{m}$	Mass flow rate	[kg/s]
r_p	Pressure ratio	[-]
T	Temperature	[^{0}C]
C_p	Specific heat at constant pressure	[kJ/kg^{0}C]
ε	Evaporative cooling effectiveness	[-]
Gh_m	Monthly sum of global irradiation	[kwh/m^2]
Gh_d	Gh_d Daily sum of global irradiation	[kwh/m^2]
Dh_d	Daily sum of diffuse irradiation	[kwh/m^2]
m^2	Square meter	[-]
T_{24}	Daily air temperature	[^{0}C]
Es_m	Monthly sum of specific electricity prod	[kWh/kW$_p$]
Es_d	Daily sum of specific electricity prod	[kWh/kW$_p$]
Et_m	Monthly sum of total electricity prod	[kWh]

E_{share}	Perceptual share of monthly electricity prod	[%]
$P\mathrm{R}$	Performance ratio	[%]

Subscripts

$a, exi\mathrm{t}$	exiting air from evaporative/fogging coolers
$aa, d\mathrm{b}$	ambient air dry bulb
C	compressor
$d\mathrm{b}$	dry bulb
$w\mathrm{b}$	wet bulb
v	water vapour
w	liquid water

CHAPTER ONE

INTRODUCTION

1 INTRODUCTION

1.1 Background

In the last three to four decades, gas turbines have become the most popular and preferred power generation plants due to the high efficiency, high power, ease of installation, and its use of cheap and clean natural gas as fuel (Chaker *et al.* 2004). Thermal efficiencies are very attractive too, ranging between 30 − 40% for simple cycle, while Combined Cycle Gas Turbine (CCGT) is in the range of 45 − 58% (Darwish *et al.* 2015). However, for utilities, co-generations and Independent Power Producers (IPPs) the challenge typically arises during the hot seasons, when both the output power and efficiency decrease especially in regions such as in the Middle East and in Sub-Saharan countries. In addition, higher electricity demand occurs when ambient temperatures are high. The ill effect of this is that for each degree rise in temperature, the output power drops between 0.54% and 0.90% (Meher-Homji & Mee III 1999), (Chaker *et al.* 2004).

One way to overcome this drop in output power is to cool the inlet air temperature. There are different types of inlet cooling technologies utilized in gas turbines such as evaporative coolers, inlet fogging, mechanical compression and absorption among others. Inlet fogging cooling is considered to be one of the most cost effective method due to it is ease of installation and operation (Kakaras *et al.* 2006). Therefore, this work presents reviews of inlet cooling technologies with the main focus on inlet fogging cooling system. A comparison of different techniques of inlet cooling is given with advantages and disadvantages of each technique and how they have been applied for gas turbines power augmentation. In addition, the work studied the gas turbine thermodynamics and the associated performance with inlet temperature.

However, inlet fogging system applies fog to provide the cooling which it evaporates in the air inlet duct of the gas turbine. The water used in this technique must be pure from ionic material and any impurities that are dissolved in water to eliminate the risk of blades corrosion or damaging of fog nozzles and possibly even compressor blading. Therefore, the water quality requirements have been considered in this thesis. In this research project the author is proposing a novel design of fogging system layout to enhance the power output of the selected Acacia power plant. This has been achieved with cooperation from the American Moistening Company (AMCO); one of the leading firms in inlet cooling.

1.2 Motivation

The motivation of this work stems from the common design methodologies that aim to produce ever more efficient power generation from gas turbines. Assuming continuous operation over many years, a one percent increase in efficiency accumulates to significant commercial benefit. The inlet cooling techniques in various work conducted have shown a dramatic increase in output power and hence increasing the gas turbines efficiency. A trend towards changing to renewable energy instead of typical fossil fuel plants is apparent, globally. However, for IPPs the challenge remains having a means to increase the output power of existing gas turbine plants as these plants may have potential years of operational lifetime remaining. Therefore, the use of inlet fogging cooling systems was investigated. However, instead of these systems being powered by the gas turbines itself as additional parasitic load, PV as a renewable energy was will be considered as the primary energy source. Thus, providing primary load to the fogging system and inherently boosting the overall existing gas turbines power output.

1.3 Problem statement

One of the challenges facing utilities, power suppliers and co-generators in hot seasons is the rapid decrease in gas turbine performance (output power and efficiency) when ambient air temperature increases. Subsequently, the cost of production per unit of energy increases. If one could increase the generation capacity of an existing gas turbine, it can deliver a direct saving on energy cost and deliver more efficient power.

1.4 Aims and objectives of this study

1.4.1 Aims

The two primary aims of this study are to:

i. give an extensive review of the different gas turbine inlet cooling technologies, and

ii. investigate the ability of the inlet fogging technique for increased power output.

1.4.2 Objectives

The objectives of the project shall be to:

i. incorporate a full layout structure of the fogging cooling system;

ii. size methodology of the PV system;

iii. evaluate the proposed system;

iv. conduct a comparison between the fogging technique and other cooling methods; and

v. finally make conclusions on the preferability, feasibility, and the limits of inlet fogging technique.

1.5 Structure of the thesis

This thesis is divided into seven chapters and is structured as follows:

Chapter 1: provides an introduction to the subject of the project. A statement of motivation, research problem and objectives is put forward. Next research outlines are presented.

Chapter 2: presents a comprehensive literature review of work that has been done on gas turbines augmentation in the past with particular attention to the inlet cooling fogging system. Next, different types of inlet cooling methods are presented. This is followed by a review on some relative case studies with inlet cooling technologies. The objective of this chapter is to provide interested readers with a general background of inlet cooling fogging systems, and their applications.

Chapter 3: begins with an introduction of simple cycle gas turbine and its components. Next, gas turbines and its cycle thermodynamics is presented. A derivation of thermal efficiency with regards to inlet temperature is delivered. Further discussion on the performance of the gas turbine with inlet temperature is presented. Gas turbine energy production share in South Africa and brief introductory to Acacia power plant is included.

Chapter 4: presents an overview of gas turbine inlet air cooling with definition of each technology. Some key benefits and drawbacks of various technologies are highlighted. Economic and environmental perspective of inlet air cooling is given. Lastly, analyses of Gas Turbine (GT) performance with ambient temperature and humidity are elaborated.

Chapter 5: illustrates in detail the state-of-art fogging system design layout. Water quality and nozzles arrangement consideration and requirements are described in this chapter. The design condition and the arrangement of the fogging system are provided. Expected impact of the cooling on the selected gas turbine is examined.

Chapter 6: In this chapter, solar PV overview and load profile analysis is presented. Solar photovoltaic system design and yield of the PV power plant is assessed. Average daily and monthly of global irradiation is obtained.

Chapter 7: is the last chapter and it represents a brief summary of the research project undertaken for this thesis. The chapter concludes the project with contributions made by this work. General recommendations are made for future work.

Appendixes:

Appendix A: Partial database of turbine inlet cooling (TIC) installations are listed

Appendix B: Illustrates global horizontal irradiation map of South Africa

Appendix C: presents psychrometric chart normal temperature in SI units

Appendix D: Weather conditions in Cape Town are tabulated in this appendix

Appendix E: Meteorological Data for Cape Town

Appendix F: Data sheet Sunmodule Plus SW 260 ploy

CHAPTER TWO

LITERATURE REVIEW

2 LITERATURE REVIEW

2.1 Background and previous research

Many studies regarding gas turbine inlet air cooling have been done over the past 30 years in an attempt to achieve higher efficiencies and enhance the performance of gas turbines generation. From cost effective methods such wetted media to refrigerated cooling and mechanical chiller for high potential cooling. Yet, the inlet fogging cooling in this particular industrial field has a lot to offer to improve and develop new techniques for better performance and high efficient gas turbine stations. This chapter intends to give a history of gas turbine inlet cooling technologies with an intention to cover the most significant and related work to inlet fogging cooling technology. Different types of inlet cooling methods are presented by different researchers. This followed by a review of some relative case studies on inlet cooling technologies. The objective of this chapter is to provide interested readers with a general background of gas turbine inlet cooling with a particular emphasis to fogging systems, and their applications.

Early papers on fog inlet cooling and wet compression appeared in the late 1940s including Kleinschmidt (Kleinschmidt 1946), and Wilcox and Trout (Wilcox & Trout 1951). Meher-Homji and Mee presented a comprehensive overview of the state-of-the-art inlet fogging system and the benefit of its application to gas turbines. Brief reviews of cooling technologies were incorporated with a main focus on direct water fogging of the gas turbine inlet air. The study assisted readers to make an assessment of the benefits that derived from applying this technology to gas turbines. Their analyses show that efficiency and power output decrease with an increase in ambient temperature. Hence, ambient temperature plays a major role on gas turbine performance (Meher-Homji & Mee III 1999).

A numerical simulation of a single shaft gas turbine utilizing two cooling methods were implemented in work done by dos Santos et al (Santos *et al.* 2012). The evaporative cooler and absorption chiller are tested and solved for different conditions. Results show that when the ambient temperature is extremely high with low relative humidity, the chiller is a more suitable cooling solution. The power output augmentation and thermal efficiency were compared. The density of the airflow is inversely proportional to the ambient temperature. If the temperature rises, the air density reduces, and that will result in less air passing through the turbine. Consequently, this results in less power generation. Moreover, the compression work increases due to the augmentation of the volume filled by the air (Santos *et al.* 2012). This is a compatible to a case where the net power output from the gas turbine is directly proportional to the air density flow, in that it decreases when ambient temperature increases (Amell & Cadavid 2002).

Ibrahim *et al* introduced a technical review for an inlet air-cooling systems that have been used mainly on gas turbine power plants. The techniques included are the Mechanical chiller, Evaporative coolers, and Absorption chillers. The diversity of the work that has been gathered in this review reflects the necessity of these techniques in improving the performance of gas turbine power output. One can conclude from this review that every technique has advantages and disadvantages all associated with effectiveness, cost, and the potential of power output increase. Results of this work show that an increase of 1^{0}C in the compressor air inlet temperature decreases the gas turbine power output by 1% (Ibrahim *et al*. 2011).

From an economical and technical perspective the benefit of fogging over traditional evaporative coolers appears to be threefold: lower capital cost, more effective cooling (ability to achieve lower gas turbine compressor inlet temperatures) and a much lower gas turbine inlet pressure drop through the application of the fogging hardware (Jones, C. and Jacobs III 2000).

Bastianen and Escue considered the water quality for evaporative inlet cooling techniques. They showed that the wetted media method cools the inlet air dry bulb temperature by as much as 90%, while fogging cools the air by up to 95%. However, because the water in the fogging system is directly evaporated in the intake air stream, it must be clear of any minerals, salts and other impurities. Therefore, in the fogging system the water is generally demineralized by reverse osmosis among other methods. They concluded that the water quality considerations are a factor that the turbine operators need to fully consider for inlet cooling. Indirect evaporative cooling seems to be the less expensive solution by taking advantage of using poor quality water. However, it is imperative that safety measures are taken to absolutely minimize the risk. Special treatments for brackish and unclear water can be achieved by chemical treatment or by reverse osmosis for direct inlet cooling (Bastianen & Escue 2009).

A case study was done by Kodituwakku on regarding the effectiveness of cooling on the Kelanitissa power station in Sri Lanka. This study implemented two approaches. Firstly, actual data acquired by the operation history of a particular plant was used to calculate the performance parameters. Secondly, the performance was analyzed for this case using thermodynamic principles. Results of the two approaches were compared. Clearly, the present performance of this power station was poor compared to designed values due to predominantly high air temperature. A 4.6% reduction in efficiency was registered at 33^{0}C (Kodituwakku 2014).

Different options for gas turbine inlet air cooling were presented (Miller *et al*. 2012). The argument was that the largest demand in Poland happens during Winter season unlike the case in Arab Gulf or Sub-Saharan regions. Considering installation of an expensive cold storage to maximize the capacity of energy production in summer peak hours is economically inefficient. Therefore, two methods were taken into account of inlet air-cooling, which were efficient in practice: evaporative cooling and chillers. The results demonstrated that evaporative cooling is a suitable solution for climates with low relative humidity, whereas in high humidity conditions, it is sensible to use chillers for cooling.

High purity demineralized water is utilized in the fogging sprayers, which is produced by RO system introduced in (Fikret YUKSEL 2015) for the single shaft gas turbine of a nominal power of 239 MW at International Organization for Standardization (ISO) conditions. The turbine has a mass air flow of 649 Kgs^{-1} while the pressure ratio is 16 bar. The fogging system consists of 5400 nozzles and operates at 70 bars. However, the literature review takes into account the effect of climatic conditions on the performance of the combined-cycle power plants. Analysis of the energy of a commercial combined-cycle gas turbine with the effects of inlet air cooling was performed. The aim of this study was to investigate the effect of the fogging system on the gas turbine system.

2.2 Different inlet cooling methods

Conditions of 15^0C and the hot summer peak periods of 40^0C, may result in a 20% drop in GT output power, whereas, if the inlet air to the gas turbine was cooled to 4^0C during these peak periods, a 27% increase may be observed.

Inlet cooling may not just enhance the power output, but it will also improve the heat rate (the ratio of fuel input rate to generated power), result in an increase in efficiency, expand the turbine life time and may delay the use for GT peak units.

Theoretically, it is possible for gas turbines to reach efficiencies of up to 65%. While, most open cycle turbines operate at around 40%. There are many ways that gas turbines can achieve higher efficiencies including increasing inlet temperatures, reducing internal losses, recycling waste heat from gas turbine exhausts and decreasing ambient temperature (Al-Ibrahim & Varnham 2010).

Different techniques are used to decrease the inlet ambient air temperature that enters a turbine:

- Wetted media evaporating cooling
- High pressure fogging

- Absorption chiller cooling
- Refrigerate cooling
- Thermal energy storage

2.2.1 Inlet fogging systems review

Gas turbine operators can opt for their method of cooling in a form of ice, fog or evaporative cooler among other methods. Inlet fogging cooling is considered to be one of the most cost effective method. Al-Ibrahim and Varnham performed a review of inlet air-cooling methods to enhance the gas turbine power generation of the Saudi electric company's during the hot seasons. Their conclusion was that the evaporative cooling system and the high pressure fogging system required a significant amount of water. Hence, it limits the use of these methods in the desert area. Despite that the absorption chiller is an expensive technique, the financial investment in it is reasonable if it is used only to improve the power in the hour's peak. Mechanical refrigeration demands large electric power from the plant itself compared to thermal energy storage methods where the requirement is lower power but needs a large storage capacity. The chosen alternative by the author was refrigeration cooling with chilled water or ice thermal storage. With this method lower inlet air temperature achieved with smaller storage size (Al-Ibrahim & Varnham 2010).

A thermodynamic assessment of inlet cooling systems for gas turbine power plants was presented in two different sites of Oman. The work was considering four techniques for inlet cooling. The techniques are evaporative cooling, fogging cooling, absorption cooling using both LiBr-H2O and aqua-ammonia, and vapor-compression cooling systems. A comparison was drawn between these techniques in relation to the electrical production augmentation, and the impact on increasing the peak capacity of the considered gas turbines (Dawoud *et al*. 2005).

In a different site, modulation and evaluation of evaporative cooling system installed in gas turbines of a combined cycle power plant in Iran was reported on by Hosseini (Hosseini *et al*. 2007). This work conducted and results showed that an increment by 14.6% for a drop in temperature from 38^{0}C with relative humidity of 8% to an air intake of about 19^{0}C.

After reviewing the prior work, the application of inlet fogging of gas turbines has seen incredible increase over the past decades yet from a technical prospective there have been few publications on this perspective. Chaker et al. approached the physical and engineering of the fogging process, droplet size measurements, droplet kinetics, and the duct behavior of the droplets in two parts. Part A (Chaker *et al*. 2002a) covers the underlying theory of droplet thermodynamics and heat transfer. This paper presented extensive results from experimental and theoretical studies with practical aspects gained in the implementation of nearly 500 inlet fogging systems. In part B (Chaker *et al.* 2002b) covers many details on nozzle technology and droplet measurements, providing experimental data on different nozzles and some recommendations for a standardized nozzle.

Bhargava et al. presents a paper that covers an analytical discussion and experimental findings of fogging nozzles characteristics related to droplets size, dynamics factors, and inlet duct configuration effects (Bharbava *et al*. 2007a). This paper is part one of a three-parts paper and providing a description of high pressure inlet evaporative fogging technology. This technique is widely used with an estimation that there are over 1000 gas turbines that currently use inlet fogging.

Part two of this paper (Bharbava *et al*. 2007b) covers the analytical and the experimental aspects of fogging presented by Meher *et al*. Their argument is that when the relative humidity increases, the effectiveness of evaporative fogging tends to be less. Overspray fogging in situations of high relative humidity can provide an intercooling effectiveness and lead to a significant power augmentation.

Part three (Bhargava *et al*. 2007c) gives general considerations for the implementation of this technology such as gas turbine icing, the design of the intake duct and in particular water quality requirements. They include practical considerations in regard to maintenance issues as well. Results of field experience related to inlet fogging are also included. It also covers a review on the status of development in the area of fogging industry. Despite the simplicity and low cost of evaporative fogging of gas turbines, however, there are several factors that play important roles to define the success of this process. It requires a complete understanding of the atomization process and critical analysis of the droplets size. Variables such as the properties of the water, the geometries of the nozzles, and the spray angle can significantly influence the cooling process.

Gas turbines have been used for power generation in several places around the world indicated in (Jaber, Q.M., Jaber, J.O. and Khawaldah 2007) and (Alhazmy & Najjar 2004). Each region has different climatic conditions. Furthermore, the periods of the peak electricity demand occur during the summer, when the ambient temperature is high. For example, in Arabian Gulf region the average ambient temperature presents a variation by more than 30^0C from summer to winter and this factor generates a large drop in output power during the summer (Nasser & El-Kalay 1991). This can pose challenges as turbine output power may vary within a matter of hours, i.e. between sunrise and sunset, yet despite variation, supply still needs to be synchronized with load or grid demand.

2.3 Case studies review on gas turbine inlet cooling

A numerical study was presented to evaluate the performance on a non-commercial single shaft gas turbine with and without cooling the intake air. Methods adopted in this study were evaporative, mechanical compression and absorption cooling system. Although, the authors have not stated how they implemented this study, the results limit the evaporative cooling effects because it depends on the ambient wet-bulb temperature. On the other hand, the chiller system offers greater cooling potential. In conclusion, the decision to choose the best cooling alternative relies on various factors such as gas turbine parameters, load type, climate conditions, cite location , power plant capacity and economic feasibility (Santos & Andrade 2012).

A thermodynamic assessment of the required power to two different inlet air-cooling systems at different location sites in Oman was presented. Extra power gain of 9.4% was obtained by evaporative cooling and 11% gain achieved with fogging for a GE Frame 6B gas turbine with rated power of 40 MW at an ambient air temperature of 48.8^{0}C. The water requirements for evaporative and fogging cooling were 12655 and 14085 tons respectively (Dawoud *et al*. 2005).

A comparison between two different air cooling system, i.e., evaporative and cooling coil is performed with a computer simulation model in order to evaluate the performance of the gas turbine unit, at Marka power station in Jordan. The authors examined the characteristics for a set of actual operational parameters including ambient temperature, turbine inlet temperature, relative humidity, pressure ratio. The results show that the evaporative cooling system enhanced the efficiency and increase the output power of the gas turbine unit in a way which was cheaper than the cooling coil system. This due to the high consumption of the coil system to run the vapor compression refrigeration unit. On the other hand, the evaporative system allows full control on the temperature inlet conditions despite of the relative humidity ratio (Jaber, Q.M., Jaber, J.O. and Khawaldah 2007).

The interesting point they made is that when the air temperature in the intake system is reduced, the net power output will be reduced as well. This is due to the fact that the utilized chiller will consume more energy to bring the temperature down.

Gas turbines, especially those are operating in an open or simple cycle, have low thermal efficiency and so the unit cost of electricity is relatively high. Many reasons can be attributed to such a low efficiency system, for example, operation method, poor maintenance, age and the engine size. Combined cycle, integration of a renewable energy source and power augmentation among other advanced technology are means to reduce the unit cost and gas emission that arise from using such a conventional system (Jaber *et al*. 2003).

Among the methods mentioned above, power augmentation serves the lowest capital cost. It is achieved by cooling the inlet air temperature of compressor intake which will increase the air density into the gas turbine and therefore, increasing the power output.

The ambient temperature strongly influences the gas turbine performance and its efficiency (Najjar 1996). Amell and Cadavid reported on their work that as the ambient temperature touches the level of 40^0C, a 25% of the rated power capacity of the gas turbine at ISO conditions is wasted. Evaporative or fogging cooling as well as mechanical cooling and thermal storage are the some of the cooling techniques. The effectiveness of the evaporative cooling method in high ambient temperature regions with low relative humidity is remarkable. When the solar irradiation is highest, i.e. typically during midday the ambient temperature is also highest and turbine efficiently lowest to inverse correlation. This correlation is very important and effectively motivates the use of solar PV electricity generation solutions in combination with turbine efficiency. Coincidentally, when there is high ambient temperature there is also large amounts of solar radiation. Thus when turbine efficiency is at its lowest, solar PV efficiency is at its highest. This allows to ways of utilizing the sun energy to supply an evaporative cooling system for gas turbine power augmentation (Amell & Cadavid 2002).

2.4 Water desalination

Water scarcity is a serious issue in many parts of the world, such as in North African countries, the Middle East, and isolated communities in dessert or arid regions in Africa. In addition, these areas are exposed to large amounts of sunlight and the rise of high ambient temperature in hot seasons. As previously discussed, inlet cooling, especially inlet fogging cooling requires water purification to protect mechanical parts of the gas turbine against metal damage, corrosion, etc. Hence, desalination techniques need to be considered and further investigated. There are a number of desalination methods proposed by researchers that are available on industry including, reverse osmosis. Many papers and practical work have appeared on RO desalination technology as this technology offers good opportunity for low cost, ease of installation and simplicity of design. There are a number of articles on solar powered RO systems as well, from pilot projects to industrial developed scale. This literature presents a number of the researches that have been done during the last 30 years in solar powered RO desalination.

2.4.1 Back ground to reverse osmosis and solar applications

Nowadays Reverse Osmosis (RO) is largely applied in the water treatment industry worldwide. These include the industrial sector as well as the civil sector. In South Africa, the production of potable water using RO is still not widely used due to several factor; such as portable water availability. The same case applies in the energy sector where the nation is heavily reliable on coal plants to generate electricity. However, in the recent years, due to low level of rain falls and scarcity of portable water, there is an urgent need to implement a new solution for this dilemma. The development of reliable RO pump application, could further help with bringing water to the people. Unfortunately, the typical areas where the use of boreholes is required for the supply of water, are also those areas with no electricity. The logical solution therefore, is a water treatment unit, powered by solar PV energy. On the other hand, utilizing inlet fogging cooling system for improved gas turbine power generation requires a standard water quality. Due to water crisis, it is recommended to withdraw water from underground resource. The associated energy consumption of pumping the water and purifying it from the same power station might not be a feasible approach. The development of solar PVRO could further enhance the performance of inlet fogging technology.

2.4.2 Solar power RO desalination

The development and execution of a RO unit powered by solar energy will not only be of great benefit for communities in remote areas; but is also seen as a cost effective of supplying portable water from brackish sources, in disadvantaged communities.

In 1997, Broker *et al.* presented a new procedure for optimum system design configuration for desalination of brackish water powered by photovoltaics. A village in Northeast Brazil was provided with the cheapest possible water supply while fulfilling the health requirements and with consideration of technical conditions. The process of RO has become increasingly important as indicated due to the low specific energy consumption of this process and advanced process made in membrane technology. The other reason is that the development in the photovoltaic field has reached a stage where solar energy can be technically and economically feasible (Broker *et al*. 1997).

In work done by Aybar et al. (Aybar *et al.* 2010) classified the RO desalination systems powered by solar energy into 3 categories: as solar thermal driven or Rankine cycle driven RO systems, PV driven RO systems and Hybrid (particularly Wind-PV). However, the choice of the system to be sued will based on the location, natural conditions, and topography of the site capacity and the size of the plant. The main selection factors may include such parameters as low maintenance, simplicity of operation, size and etc. This paper proposed small PV driven RO systems for small farmer's villages with few populations in rural areas of Central Asia. This system is cited as being suitable for drinking water (Aybar *et al.* 2010).

2.5 Summary

In this chapter literature review of previous work on gas turbine inlet air cooling and its application were expounded on. In the beginning of the chapter, the literature overviews inlet air cooling from economical and technical perspectives as well as an overview of reverse osmosis, water quality consideration and solar applications. The effects of inlet cooling on gas turbines generation were discussed. The objective of this chapter is to provide a general background of Inlet air cooling with an emphasis on inlet fogging cooling applications.

CHAPTER THREE

GAS TURBINE THERMODYNAMICS

3 GAS TURBINE THERMODYNAMICS

3.1 Introduction

A gas turbine is usually classified as a heat heavy frame industrial machine that uses high-temperature, high-pressure gas to convert chemical energy to mechanical work. Gas turbines are also referred to as (combustion turbines) CT because of its function to burning a fuel in air. The result is a high temperature gas. Gas turbines are a quick and fast in operation and therefore, they are ideal for peak-load and emergency applications for short periods. It gained widespread popularity due to their high efficiency, robustness, lightweight, and its use of cheap and clean natural gas as fuel. The capacity size ranges from micro turbines to 500 MW. Gas turbine fuel is either Natural Gas (NG) or Diesel and sometimes uses Heavy Fuel Oil (HFO) (Kodituwakku 2014). The efficiency of GT is in the range of 30-36%, and GTCC is in the range of 45-58% (Darwish *et al*. 2015). A typical gas turbine consists of three main components namely a compressor, a combustion chamber and a turbine. Figure 3.1 illustrates the main components of a gas turbine.

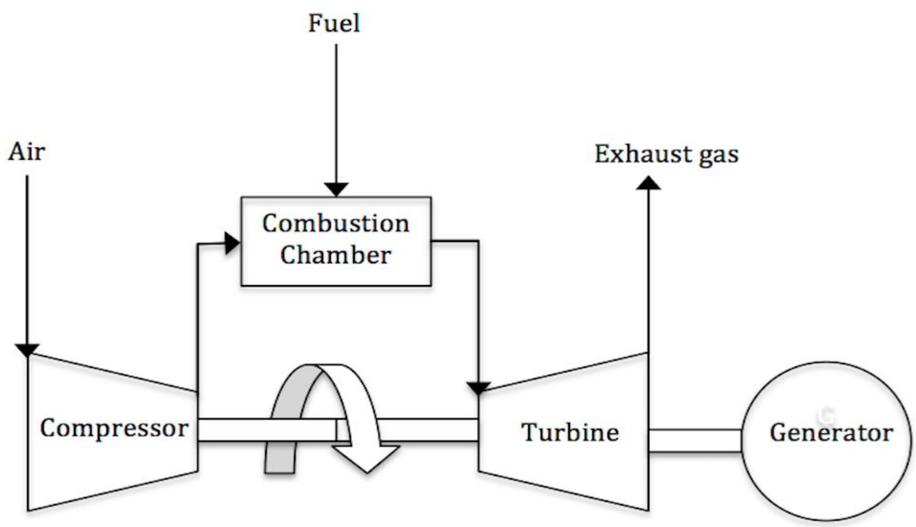


Figure 3.1: Simple layout of a gas turbine

The mechanism of the gas turbine starts when air drawn is in through the end of the turbine and the compressor section of the turbine compresses the air together. As the air is compressed it gets hot and pressure increases. Next, the fuel is injected into the combustor where it mixes with a hot compressed air and it burns in the combustion room. This is pure chemical energy at work. The hot gas created from the ignited mixture moves through the turbine blades forcing them to spin with a fixed speed. Chemical energy has been converted into mechanical energy. The turbine then captures the energy from the expanding gas that causes the drive shaft which connected to the generator to rotate. The rotor part of the generator is surrounded by coils of copper wire, thus when it rotates it creates a magnetic field that lines up electrons around the coils and causes them to move inside. This is how electricity is generated at the terminal part of the generator.

3.2 GT components

Gas turbines operate according to the Brayton cycle and consists of the following components:

3.2.1 Intake

This is the place where air enters the compressor. It has an air filter to prevent dust from entering the machine when the air passes through it. The design of the inlet duct in front of the compressor works as a diffuser which makes the air decelerates and converts part of the air's kinetic energy into pressure. Figure 3.2 shows an air filter attached to the air inlet to the compressor. The process of the compression is volumetrically constrained. Therefore, if the air ambient temperature increases, the air density and mass flow decreases, and that would increase the specific power consumed by the compressor. Consequently, this decreases the GT power output.

Figure 3.2: Inlet air of APR gas turbine (APR, E., FT8)

Figure 3.3 shows the effect of compressor inlet temperature on the GT output and heat rate.

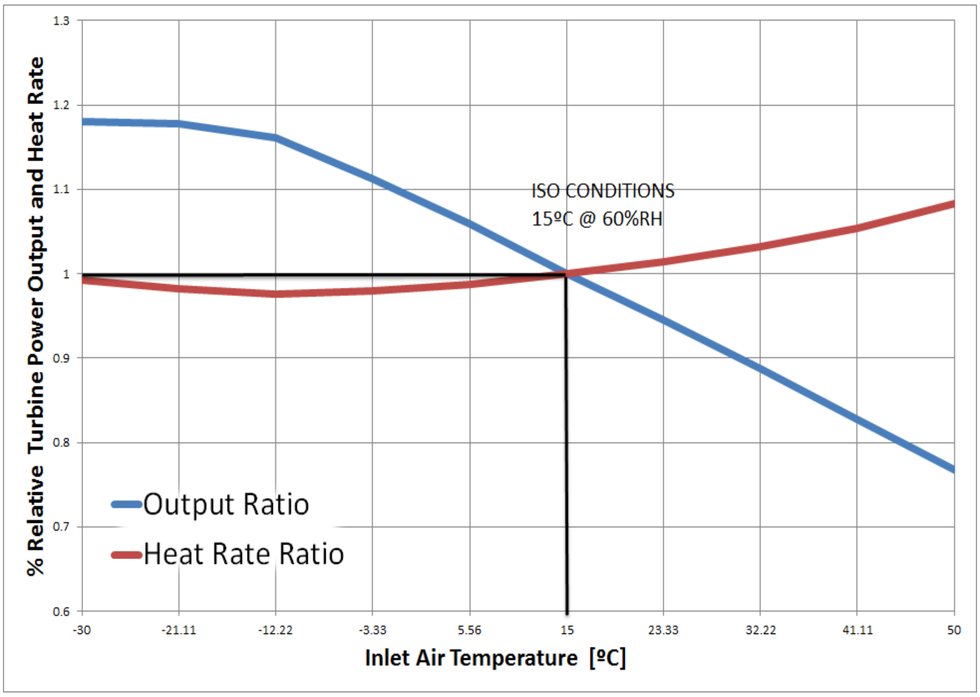


Figure 3.3: Typical inlet air-cooling impacts on combustion turbine performance.

Different means are used to reduce the air ambient temperature either by evaporative cooling, fogging, or chilled water system.

3.2.2 GT compressor

The main features of the compressor are the pressure ratio (rp), consumed power, flow rate volume, and the permissible shaft length. They are three types of compressors used in GT systems: axial, centrifugal, and a combination of both. Axial compressors have a longer shaft length than centrifugal ones. Therefore, an axial compressor handles a much wider range of volume flows, and is used in heavy utility gas turbines. On the other hand, centrifugal compressors have small-size, short shafts, and is used only in small gas turbines.

3.2.3 GT combustor

The air leaving the compressor enters the combustion chamber, called the combustor, where fuel is injected. The function of the combustor is to convert fuel chemical energy to thermal energy. Different classes of the combustors:

- Annular (continuous chamber that encircles the air in a plane perpendicular to the air flow)

- Can-annular (similar to the annular but incorporates several can-shaped combustion chambers rather than a single continuous chamber)

- Silo (silo, frame-type, combustor has one combustion chamber mounted externally to the gas turbine body) (Darwish *et al.* 2015)

3.2.4 Gas turbine

A simple gas turbine cycle consists of a compressor, turbine, and generator usually mounted on a single shaft and combustion chamber. When started, the generator is usually operated as a motor to get sufficient rotor speed. Then, the gas turbine is ignited, and power supply to the generator-motor is switched off. The GT accelerates until it reaches its nominal speed, and generator is synchronized and connected to the power grid. The hot gases produced in the combustor are expanded in the turbine to provide the mechanical energy that operates the compressor, and the balance produces the electric power.

3.3 Gas turbine associated performance with inlet temperature

A review of the thermodynamics of gas turbine cycles could help to gain a better understanding of what affects the power augmentation of gas turbine and its dependency on inlet temperature.

3.3.1 Gas turbine cycle

The pressure volume (P-V) diagram shown in Figure 3.4 demonstrates the gas turbine cycle. Process 1-2 represents compression, 2-3 represents heat addition in the combustor, and 3-4 represents the expansion in the turbine. The dash dots represent the work of the cycle.

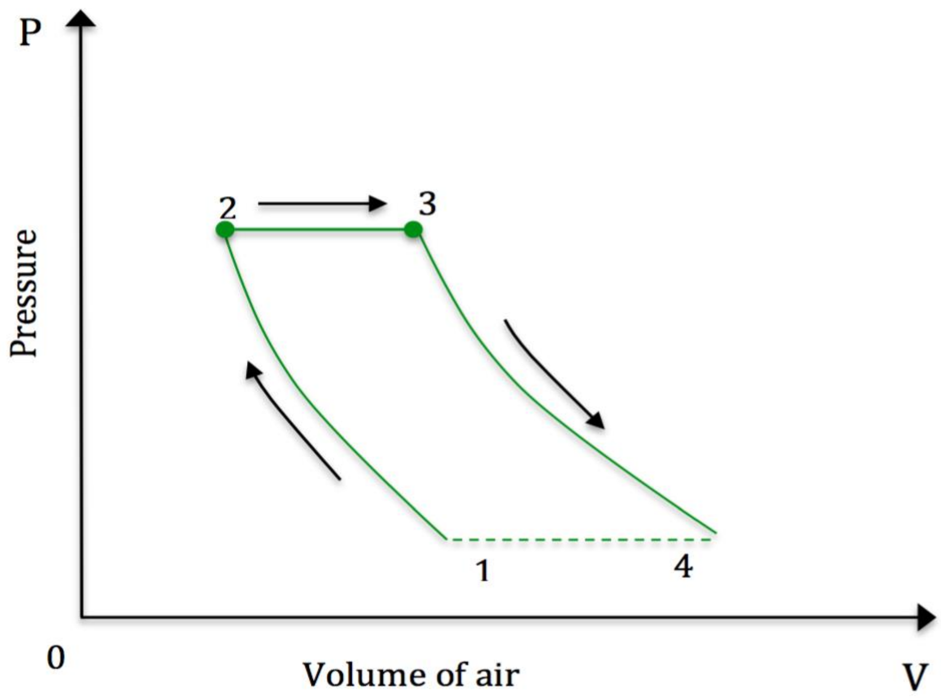


Figure 3.4: Typical P-V diagram.

An entropy diagram is an alternate way to present the gas turbine cycle and it is shown in Figure 3.5. $1-2$ represents the ideal compression process, with $1-2'$ being the actual compression. Heat addition in the combustor is represented in line $2'-3$. Line $3-4$ represents the expansion in the turbine with the real expansion being $3-4'$. The formula $C_P[T_2'-T_1]$ is given for compression work and the work of expansion is given by $C_P[T_3-T_4']$. Gas turbines can produce output power when the work of expansion is greater than the work of compression. In other words, the line indicated by $3-4$ must be greater than $1-2$ (Meher-Homji & Mee III 1999). The compression process consumes a large percentage of the total work produced by gas turbine. Any means of reducing the work of the compression will therefore, enhance the power output of the gas turbine.

If steady-state with no kinetic or potential energy change, each step could be expressed as

$$q = w + h_e - h_i \tag{3.1}$$

Work input occurs in step $1-2$ (i.e. it requires power) while work output occurs in step $3-4$ (that's the turbine work).

$$W_{net} = w_{12} + w_{34} = (h_1 - h_2) + (h_3 - h_4) \tag{3.2}$$

Heat transfer happens in step $2-3$, so that's the enthalpy change from $2-3$

$$q_{in} + q_H = q_{23} = h_3 - h_2 \tag{3.3}$$

Therefore, the thermal efficiency is

$$\eta_{th} = \frac{(h_1 - h_2) + (h_3 - h_4)}{h_3 - h_2} = \frac{w_{net}}{q_{in}} \tag{3.4}$$

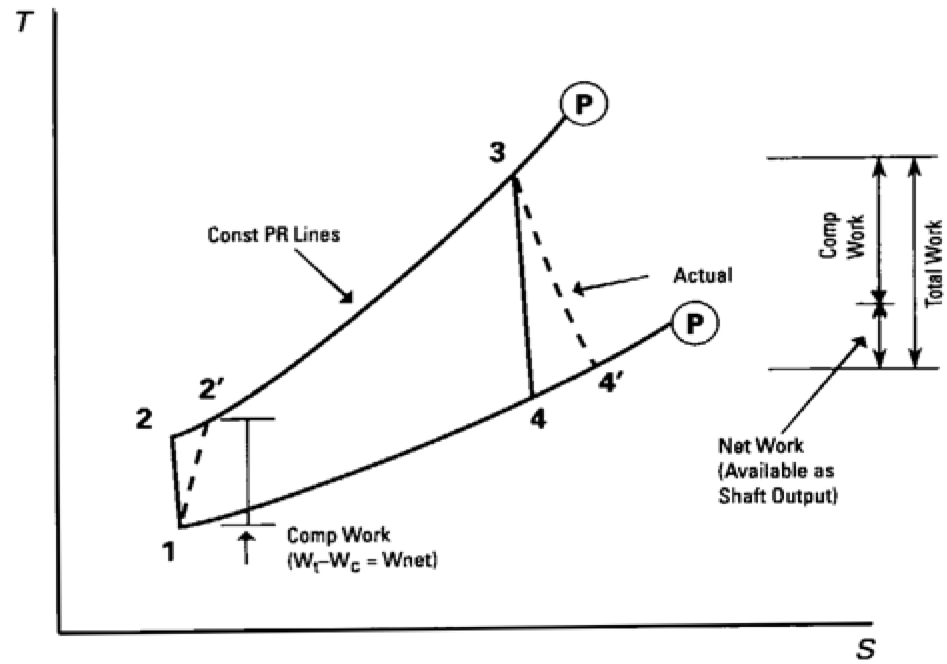


Figure 3.5: Temperature entropy diagram (T-S) (Meher-Homji & Mee III 1999)

The compressor work is given by:

$$\left(\frac{W}{J}\right)_{Compr} = \frac{h_2' - h_1}{\eta_c} = \frac{C_p[T_2' - T_1]}{\eta_c} = \frac{C_p T_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{\eta_c} \qquad (3.5)$$

It can be seen that increasing T_1 would increase the compression work. Gas turbine thermal efficiency can be reduced to the equation 3.6

$$\eta_{th} = \frac{\frac{\eta_t T_3}{B} - \frac{T_1}{\eta_c}}{\frac{T_3 - T_1}{B - 1} - \frac{T_1}{\eta_c}} \qquad (3.6)$$

Where

$$B = (P_2/P_1)^{\frac{\gamma-1}{\gamma}}$$

η_c= Compressor efficiency

η_t= Turbine efficiency

T_3= Turbine inlet temperature

P_2, P_1= Cycle pressure limits

It can be seen that the cycle efficiency decreases with the increase in compressor inlet temperature. The effect of increasing the inlet temperature, the compressor discharge pressure and temperature decreases and more fuel is required to reach the same Turbine Inlet Temperature (TIT). As the compressor work decreases, the output work decreases as well.

3.3.2 Impact of ambient temperature on air density

Equation 3.7 describes the relation of air density with temperature

$$\rho = \frac{P \times 144}{RT} \tag{3.7}$$

Where

P= Pressure
R= 53.3
T= Temperature ^{0}C
ρ= Density, kgm^{-3}

Rising temperature causes a fall in air density as illustrated in Figure 3.6, and this causes a reduction in the mass flow rate through the gas turbine. The speed of a single shaft gas turbine is essentially restricted to volumetric flow rate intake. The amount of power produced is limited by the fuel flow, which in turn affects the turbine inlet temperature.

Equation 3.8 states that the mass flow is proportional to the pressure of compressor inlet and inversely proportional to the inlet temperature. The mass flow is also proportional to the absolute pressure (P_3) at the inlet nozzle. This makes it inversely proportional to the square root of the turbine inlet temperature (T_3) (Meher-Homji & Mee III 1999).

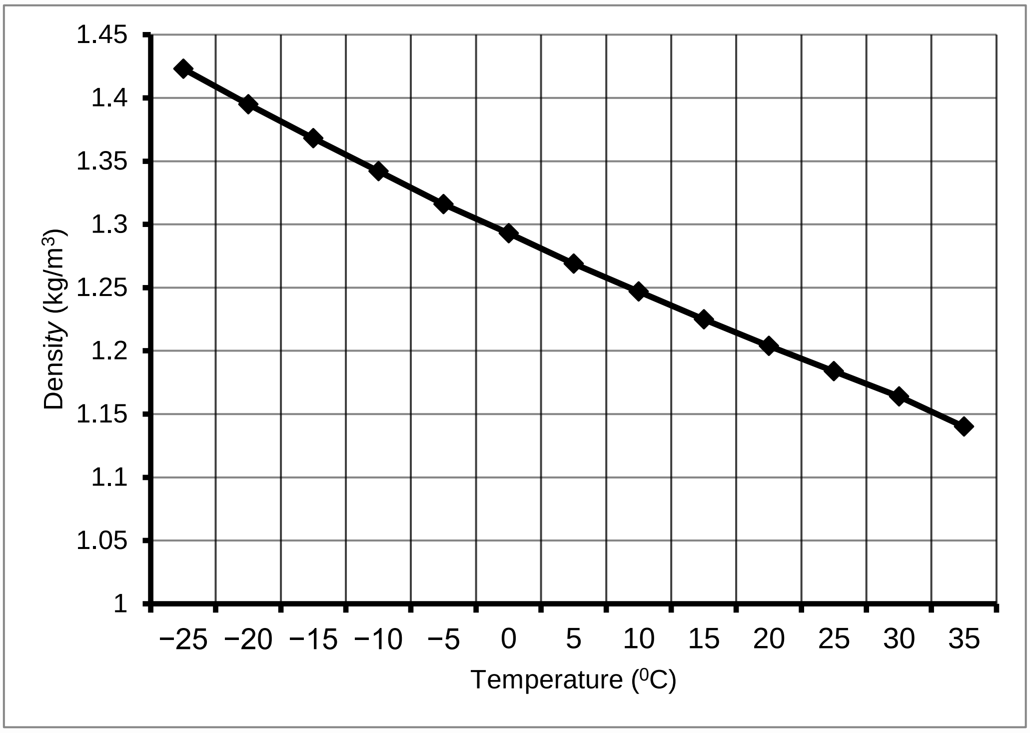


Figure 3.6: Ambient temperature vs. air density

$$m = K_1 \frac{P_1}{T_1} = K_2 \frac{P_3}{\sqrt{T}_3} \tag{3.8}$$

3.3.3 Ambient temperature effect

As has been sated earlier, with every degree rise in temperature, the power output drops by between 0.54% and 0.9%. This effect is the focus of this thesis. As the temperature decreases the air becomes denser. Therefore, the turbine will operate at higher mass flow rate and higher pressure ratio. Consequently, this increases power output and improved heat rate.

$$m = K_1 \frac{P_1}{T_1}$$

The equation for thermal efficiency for an ideal Brayton cycle shall be derived, assuming air-cold standard analysis. It is a simple expression and it is a function of pressure ratio and *K* which is typically 1.4. The equation for thermal efficiency is

$$\eta = \frac{w_{net}}{q_{in}} = \frac{q_{net}}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}} \quad (3.9)$$

or

$$\eta = 1 - \frac{q_{out}}{q_{in}} \quad (3.10)$$

From entropy diagram

$$\eta = 1 - \frac{h_4 - h_1}{h_3 - h_2} \quad (3.11)$$

And that equals to

$$= 1 - \frac{C_p(T_4 - T_1)}{C_p(T_3 - T_2)} \quad (3.12)$$

Making the use of cold-air standard

$$\eta = 1 - \frac{T_1(\frac{T_4}{T_1} - 1)}{T_2(\frac{T_3}{T_2} - 1)} \quad (3.13)$$

From ideal Brayton cycle

$$\frac{T_2}{T_1} = (\frac{P_2}{P_1})^{\frac{k-1}{k}} \quad (3.14)$$

And

$$\frac{T_3}{T_2} = \frac{T_3}{T_4} \cdot \frac{T_4}{T_1} \cdot \frac{T_1}{T_2} \quad (3.15)$$

The product of $[\frac{T_3}{T_4} \cdot \frac{T_1}{T_2}]$ is equal to 1

Hence,

$$\frac{T_3}{T_2} = \frac{T_4}{T_1} \quad (3.16)$$

And from equation 3.13

$$\eta = 1 - \frac{T_1}{T_2} \quad (3.17)$$

At the end

$$\eta = 1 - \frac{1}{(\frac{P_2}{P_1})^{\frac{k-1}{k}}} \quad (3.18)$$

Where $rp = \frac{P_2}{P_1} = \frac{P_3}{P_4}$ is the pressure ratio.

It can be noted that the efficiency also depends on T_3, turbine inlet temperature, pressure ratio rp, and $K = \frac{C_p}{C_v} = 1.4$. Gas turbine thermal efficiency increases if the pressure ratio increases. Typical values of rp are between 5 and 19. The ratio can further be exceeded. However, the cost of construction makes it impractical and high values can decrease the operation of the compressor.

3.4 Energy in South Africa

South Africa (SA) is by far the largest in Africa in terms of electric power production with approximate capacity of 250000 GWh annually. Most of this power is consumed domestically and around 12000 GWh are exported to neighbouring countries in the act of participating in the Southern African Power Pool (SAPP). Eskom is the national utility power supplier for electricity in SA. Most of the power stations in SA are owned and managed by Eskom, where generation accounts for 95% of all the electricity in SA and 45% of all electricity produced on the African continent. It uses a mix of energy source of coal, oil, natural gas, nuclear energy, hydro and renewable energy. Figure 3.7 shows the energy consumption mix in SA. Coal is the dominant fuel source, being responsible for 70% of the country’s power production in 2014. The other sources comprise the following share:

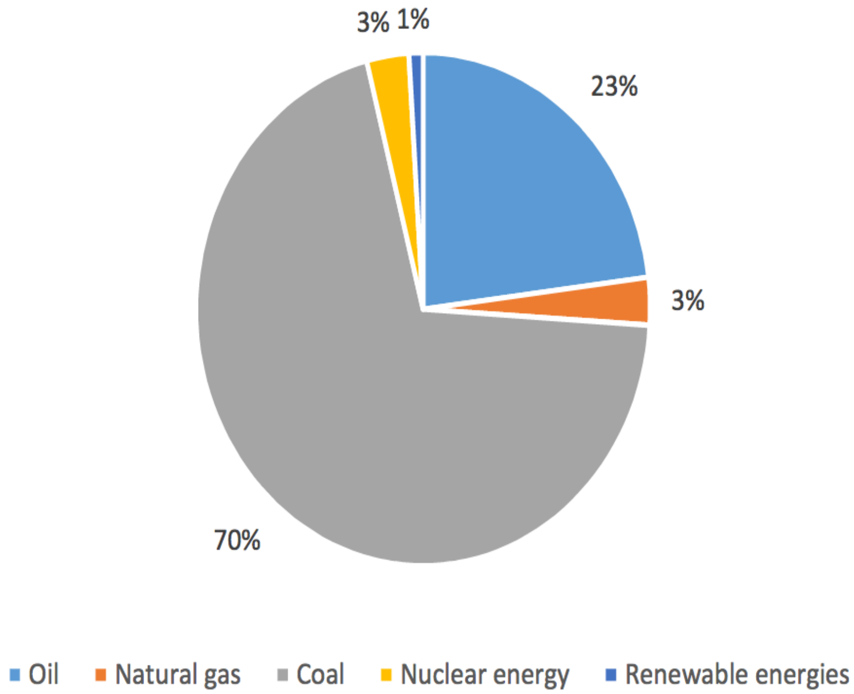


Figure 3.7: Energy consumption mix in South Africa (Source: DMR, 2015)

Table 3-1 provides a list of maximum electricity generating capacities per fuel source available to the national grid, i.e. the maximum power the power station can deliver to the grid.

Table 3.1: Maximum available energy resources to the national grid per fuel source

Type	Capacity [MW]
Coal	40036
Gas turbine	3449
Hydro	3573
Nuclear	1860
Wind	1369
Solar PV	1149

Solar CSP	200
Landfill	7.5
Imported Hydro	1500

3.4.1 GT in South Africa

As indicated previously, the coal accounts for 70% of the primary energy consumption, while the natural gas share is 3% of all. Table 3.2 presents a list of gas turbine stations capacity, location and the current status in South Africa.

Table 3.2: Gas turbine power plants in South Africa (source: www.eskom.co.za)

Power plant	Province	Date commissioned	Installed capacity [MW]	Status	Coordinates
Acacia Power Station	Western Cape	1976	171	Operational	33°53′00″S 18°32′08″E
Ankerling Power Station	Western Cape	2007	1338	Operational	33°35′32″S 18°27′37″E
Gourikwa Power Station	Western Cape	2007	746	Operational	34°10′00″S 21°57′38″E
Newcastle Cogneration Plant	KwaZulu-Natal	2007	18	Operational	27°47′08″S 29°58′11″E
Port Rex Power Station	Eastern Cape	1976	171	Operational	33°01′43″S 27°52′52″E

Avon Peaking Power	KwaZulu-Natal	2016	670	Operational	29°25′10″S 31°09′41″E
Dedisa Peaking Power	Eastern Cape	2105	335	Operational	33°44′33″S 25°40′22″E

3.4.2 Acacia power plant

Acacia power plant is owned by Eskom and it is part of Eskom’s peaking power stations. Acacia is located on the outskirts of the city of Cape Town. Situated at the southern end of the national grid, Acacia also operates predominantly in the synchronous condenser mode to regulate voltage. In addition, it provides an off-site electrical supply to Koeberg nuclear power station (Eskom 2014).

Figure 3.8: View of Acacia power plant (Source: Google Maps)

Unlike conventional gas turbines, Acacia is equipped to burn a variety of fuels such as oil and gas. This is an advantage of the modular design, for routine and planned maintenance can be done considerably faster.

Acacia has three gas turbine generators, which are driven by engines similar to those of Boeing 707 aircraft. It was commissioned in 1976. Each unit is capable of delivering an output of 57 MW at base load. However, the peak output is designed to be 60.8 MW at atmospheric condition. The peak load output can only be sustained for three hours. This gives a total installed capacity of 171 MW. The TP4 Twin Pacs (as the sets are called) consists of three primary units illustrated in Figure 3.9:

- The gas turbine unit
- The generation unit
- The control unit

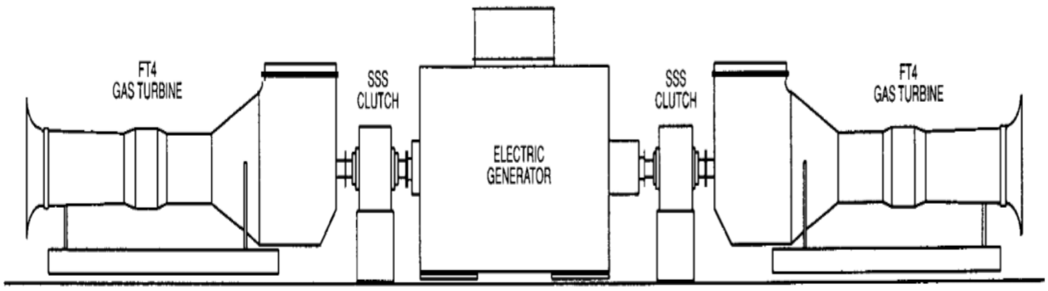


Figure 3.9: Standard installation of TP4 Twins Pac (Backus 1989)

The turbine and the generator units consists of two gas turbines/engines connected through self-shifting clutches to an electric generator with a through-drive connected brushless exciter. Adjacent to the turbine/generator is the control unit, which contains all of the controls and instruments necessary to operate the power plant. The entire unit occupies approximately 48.9m × 12.2m of space with the turbine-generator module on a common 37.25m×7.23m reinforced concrete pad.

The gas turbines run on Kerosene and each unit consumes 5.7 liter/sec at a base load of 57 MW. The overall thermal efficiency of the unit is 28.8%.

3.5 Summary

In this chapter an overview of a simple cycle gas turbines and its components are provided. Gas turbine associated performance with inlet temperature was also discussed in detail to elaborate the effects of inlet cooling on power gas turbines augmentation. A thermal efficiency with regards to inlet temperature was delivered. This information is intended to give a complete understanding of gas turbines thermodynamics and its association with ambient air temperature. This chapter has included gas turbines energy contribution in South Africa and brief introductory to Acacia power plant.

CHAPTER FOUR

OVERVIEW OF GAS TURBINE INLET AIR COOLING TECHNOLOGIES

4 OVERVIEW OF GAS TURBINE INLET AIR COOLING TECHNOLOGIES

4.1 Introduction

Cooling off and releasing some energy is a strategy among humans in summer days. Gas turbines in this manner are just like humans on high temperature days. They need to be cooled down. Instead of choosing what is available between pool, sea or ocean, turbine operators can opt for their method of cooling in a form of ice, fog or evaporative cooler among other methods. Without a doubt, cooling inlet air is an economical approach for existing gas turbines.

Taking advantage of the dependency of combustion turbine's output on the air mass flow through the compressor, and since the intake is strict in volume due to the turbine design, reducing the air temperature increases the air density and, hence, the output power increases.

The matter may seem simple, however, for operators the question is not whether to use an inlet cooling system, but which technology best meets the requirements. Various factors to consider for different environments include annual local temperature, relative humidity, water availability, construction cost, operation cost and periodic maintenance.

In this chapter, inlet air-cooling technologies will be reviewed in detail. This includes a definition of every available technology considering the major factors that play a significant role to make best decision for operators at different conditions with a comparison between the technologies. In addition, a critical review of previous publications on the field will be discussed given some real case studies and other industrial application reviews.

4.2 A review of GT inlet air-cooling technologies

Turbine inlet air-cooling employs different technologies including the cooling down of the air intake of the combustion. This directly yields to a mass increase of air and a decrease of the compressor's work. Consequently, it also gives rise to power augmentation and overall a better performance. This technology has been widely used for different range of gas turbine power stations in hot climates with high ambient temperatures at peak demand periods.

Various technologies are available in the industry market (CB Meher-Homji 1983). Each particular technology has its benefits and drawbacks. Generally, they are classified into three categories (Meher-Homji & Mee III 1999):

- Evaporative methods: This is through either *Wetted media* and *High-pressure fogging*
- Refrigerated inlet cooling: This includes *Absorption* or *Mechanical refrigeration*
- Thermal energy storage system: Utilizing off-peak hours to produce *Ice storage* or *Chilled water storage* used to cool the inlet air during hot hours of the day

All of the technologies listed above have inherent pros and cons. Many published articles are presented on these technologies. A number of these publications were reviewed for the literature review. In the next section, definitions and recent updates of each technology shall be dealt with.

4.3 Definitions

Gas turbine air inlet is introduced first to define the available inlet air-cooling techniques.

4.3.1 Gas turbine air inlet

The incoming air to the combustion turbine first goes through a weather hood to a filter housing. The weather hoods protect the filters from the effects of rain and sun and help minimize the amount of debris drawn into the filters. The air is drawn into through the filter canisters. These filters remove particles from the incoming air before it is admitted to the compressor inlet. The filter banks are constructed with self-cleaning filter assemblies. The pulse filter cleaning system uses a reverse airflow technique to clean debris from in-service filters. From the filters the air goes to an evaporative cooling media section. The evaporative cooler utilizes water to cool the incoming air. The cooler air results in a greater density of mass flow. This maximizes the gas turbine's power output when operating during hot days. Downstream of the evaporative cooling section are the mist eliminators. The mist eliminators capture loose droplets of moisture that may be carried out of the evaporative cooler media. The inlet air flows through the two banks of filters and combines in the inlet manifold. From the inlet manifold the air is drawn through silencers and then directed in the axial flow compressor inlet.

4.3.2 High-pressure fogging

It is a spraying mechanism of fine droplets between 5-20 microns in diameter into the gas turbine intake at pressure 70-200 bar (Meher-Homji & Mee III 1999). A simple fog system consists of high-pressure pumps, Programmable Logic Controller (PLC) and an array of fog nozzles mounted in the inlet air duct. Fogging requires demineralized water to be used for inlet air cooling to prevent fouling of the compressor blades and eliminate possible damage to the fog nozzles that would occur if water was not demineralized. Figure 4.1 illustrates a perspective layout of GT with high pressure fogging system. This technology will have a detailed description in the next chapter.

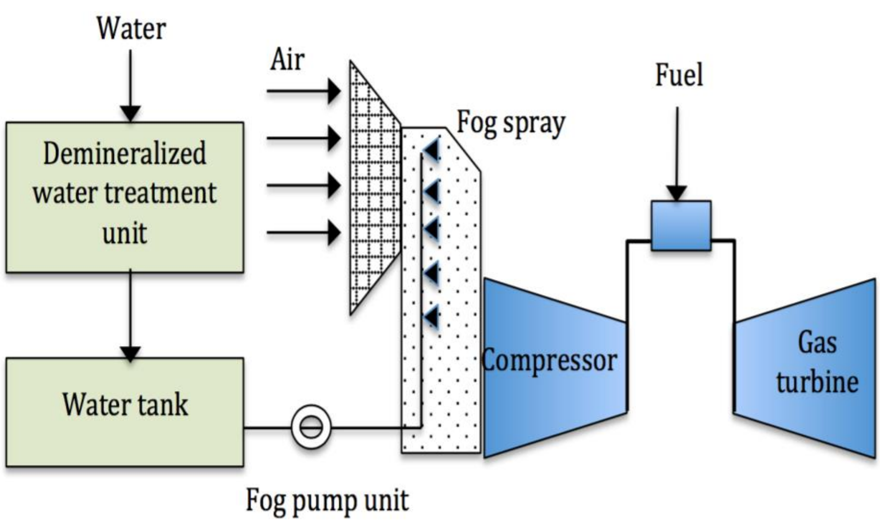


Figure 4.1: Inlet air cooling with high pressure fogging

4.3.3 Wetted media

This method uses a wetted rigid media where water is distributed throughout the header and where air passes through the wet porous surface. Part of the water is evaporated, absorbing the heat from the air and increasing its relative humidity. It is considered to be the most appropriate cooling system in hot dry areas, because it utilize the latent heat of vaporization to cool ambient temperature from the dry-bulb to the wet-bulb temperature (Al-Ibrahim & Varnham 2010). A simple system of wetted media is illustrated in Figure 4.2.

4.3.4 Absorption

The method is shown schematically in Figure 4.3. This application is used in areas where relative humidity is high. It takes the turbine exhaust gases to produce chilled water in a double effect lithium-bromide absorption chiller. The chilled water is passed through a heat exchanger to cool the ambient air temperature. This system has the advantage of increasing the gas turbines performance better than evaporative cooling. However, it requires a higher capital cost and high O&M cost.

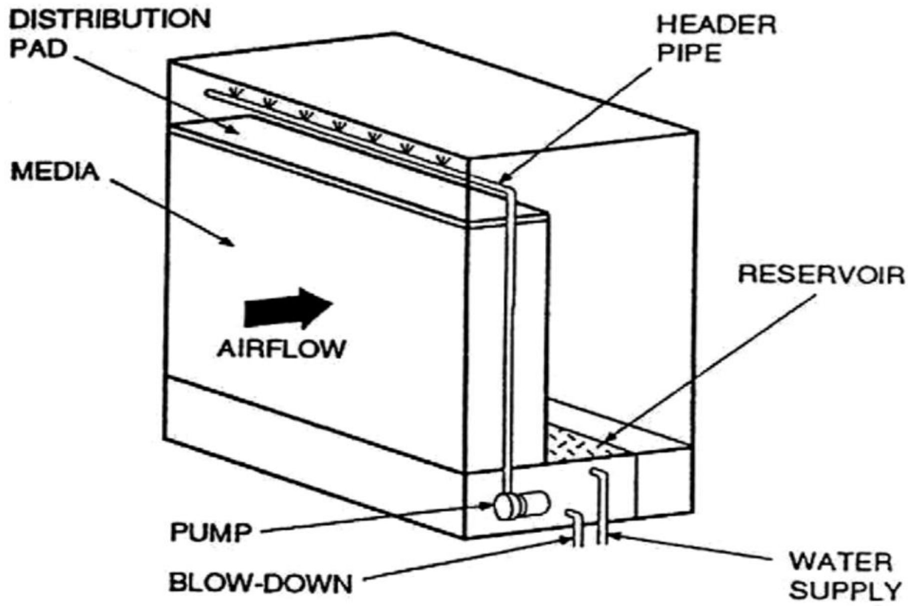


Figure 4.2: Schematic of the wetted media evaporative cooler (Hosseini *et al.* 2007)

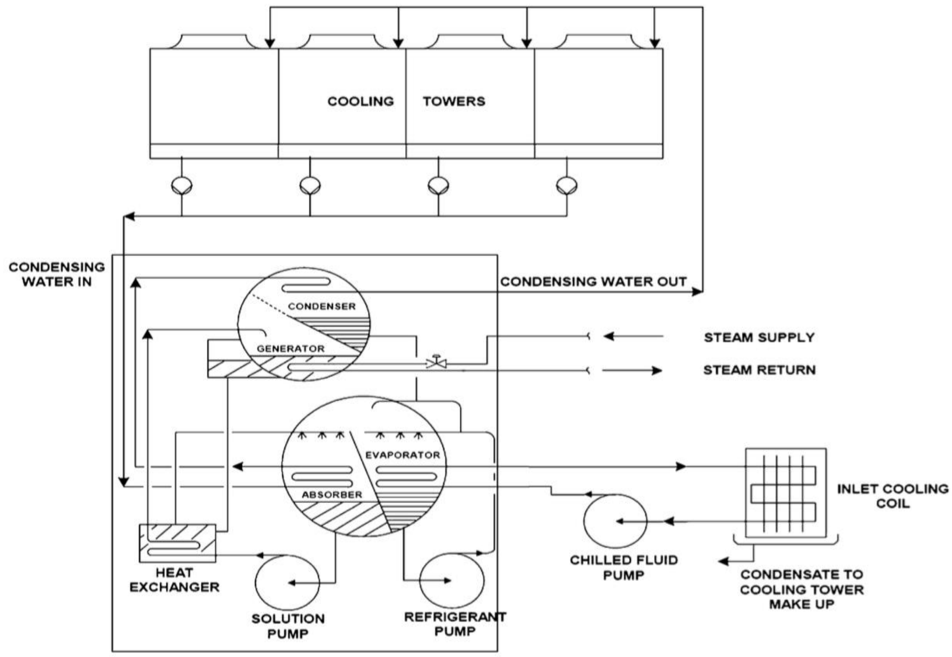


Figure 4.3: Schematic layout of absorption chiller (Darwish *et al*. 2008)

4.3.5 Mechanical refrigeration

Mechanical refrigeration is a process by which heat is removed from a location using a man-made heat-exchange system. Figure 4.4 shows mechanical refrigeration system. With this technology the air temperature can be attained to any desirable degree irrespective of ambient temperature and relative humidity. On the other hand, this system has high initial capital cost compared to evaporative cooling techniques.

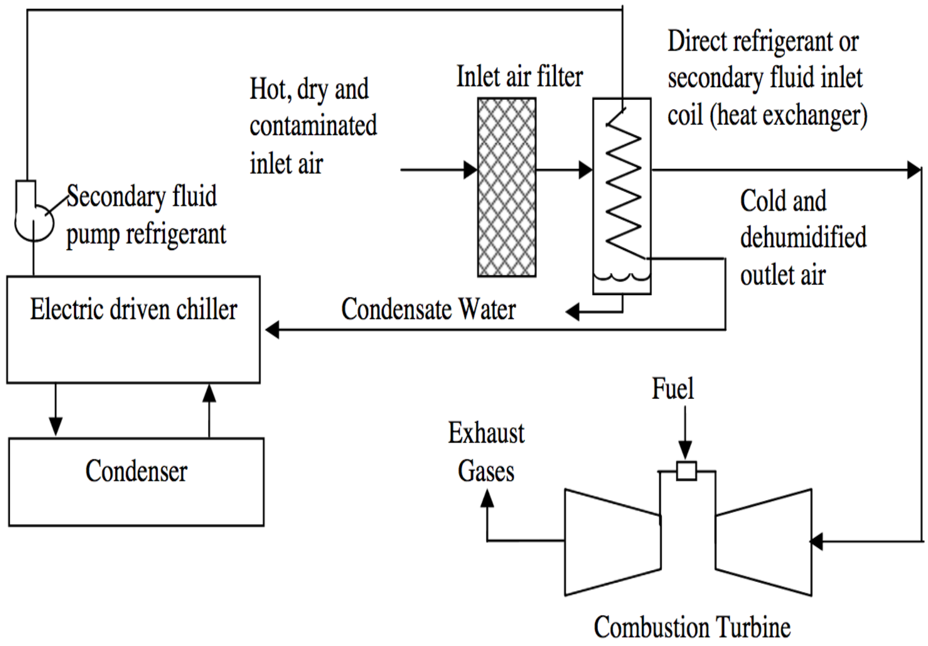


Figure 4.4: Mechanical refrigeration (Abdalla & Adam 2006)

4.3.6 Thermal energy storage

A thermal energy storage tank is a thermal accumulator that allows the storage of chilled water or ice produced during off-peak periods to chill the turbine inlet air during high-peak hours to boost the power output. A thermal energy storage tank reduces operational cost and refrigerant plant capacity. The production of the storage (chilled water or ice) has the advantage of using cheaper tariffs when demand is low, which usually coincides at night to enhance the gas turbine performance during peak hours of hot days. Figure 4.5 illustrates a thermal energy plant for air inlet cooling.

4.4 The history of inlet air cooling technologies

The first technology of inlet air-cooling made its appearance since the forties of the last century (Cyrus B. Meher-Homji 2000). Early investigations on wet compression were done by (Kleinschmidt 1946) followed by a report by Wilcox and Trout (Wilcox & Trout 1951).

At later stage, high-pressure water fogging gained a popular renown in the industry and it is being applied in gas turbine augmentation. The first report on this concept was published by (Nolan, J.P. and Twombly 1990). Meher-Homji and Mee published a comprehensive review on inlet cooling technology in 1999 (Meher-Homji & Mee III 1999).

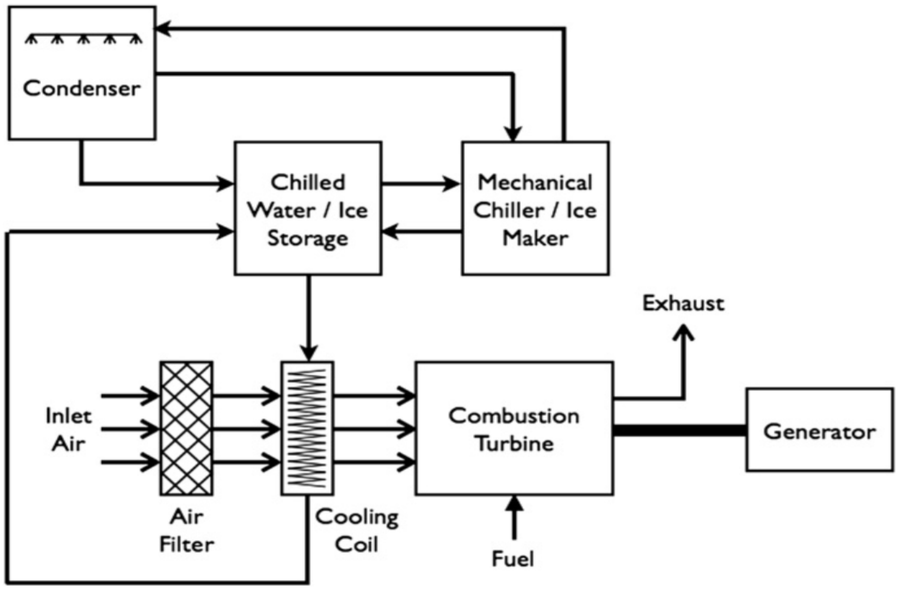


Figure 4.5: Thermal energy storage for air inlet cooling (Al-Ibrahim & Varnham 2010)

Bacigalupo *et al* covered a detailed discussion of cooling technologies in 1993 (Bacigalupo, E., Tasso, L. and Zinnari 1993). Among other technology, the first application of ice storage was presented for a system in Lincoln, Nebraska USA in 1992, and another direct air-cooling system for a plant in Battle Greek, Michigan in the USA in 1988 (MacCracken 1994)

No particular technology can be considered the best option. Although, the focus of this work is on high pressure fogging technology which is considered the most cost effective technology. Table 4.1 presents several available technologies for cooling the air entering a CT, and highlighted the major benefits and drawbacks of each technology.

Table 4.1: Pros and cons of different technologies available for CTIAC (Al-Ibrahim & Varnham 2010), (Omidvar 2001) and (Andrepont 2001).

Technology	Pros	Cons
High pressure fogging	• Allows fine control of temperature over wider range of conditions than evaporative cooling • Low capital and operational cost • Simple, easy to install and operate • Low maintenance required • Low parasitic power consumption • Unlimited time on operation • Further power gain by overspray fogging	• Undesirable at location with water scarcity • Limited power augmentation • Requires demineralized water • Can not quite match the amount of cooling provided by a chiller
Evaporative cooling	• Instant on and off operation • Low capital cost • Uncomplicated and reliable system	• It is limited in the amount of cooling it can provide • Not suitable for hot humid conditions • Cost can increase if there is not adequate space
Absorption chiller	• Can induce any desired amount of cooling • Not sensitive to ambient air wet bulb temperature • Low electrical parasitic load	• Has much higher initial and operating cost • Longer delivery and installation time
Mechanical refrigeration	• Better increased performance compared to evaporative and fogging • Unlimited operational time • Not affected by humid environments	• Expensive capital cost • Large power demand • High O&M cost • Long delivery and installation time •
Mechanical refrigera-	• Higher potential of power boost than	• Relatively high cost

tion (ice storage)	evaporative or fogging • Better effectiveness in humid conditions • Can benefit of low night-time tariff to produce and store ice for peak hours operation	• Requires addition storage volume • Limited period of operation • Complex system requires O&M expertise • Long delivery and installation time • Requires time to make the ice
Mechanical refrigeration (chilled water storage)	• Requires low electric power during peak hours • Simple and reliable system • Enhance greater performance compared to evaporative and fogging	• Requires large amount of water and storage volume • Limitation of the cooling temperature • Limited operation time per day

4.5 Advantages and disadvantages of inlet cooling

GTIC is used by thousands of GT-based power plants all over the world to boost the power output capacity when it is most needed, and when it is also most valuable. This trend is seen not particularly in the United State but also in some countries in the Middle East and Africa. This is due to the higher growth demand in power, which is directly linked to 40% of power usage of air conditioning (Andrepont *et al*. 2012).

GTIC can provide several economic and environmental benefits for power producers as follows:

4.5.1 Economic benefits

- Boosts power output when most needed
- Reduces capital cost for increased capacity
- Increases GT fuel efficiency by lowering heat rate

4.5.2 Environmental benefits

- Reduces emissions of pollutants

- Reduces emissions of global warming gas CO_2

- Eliminates setting up new power plants

On the other hand, the drawback of inlet cooling is the capital cost and it requires additional maintenance cost for its equipment.

4.6 Analysis of GT performance with ambient temperature and humidity

This section provides a detailed discussion on the performance of GT_S and its dependency on ambient air temperature. The rated capacity of all gas turbines are based on ISO standard ambient air at 15^0C, 60% relative humidity, 1 bar at sea level and zero inlet and exhaust pressure drops. In this section, a numerical simulation is presented for a single shaft gas turbine. The specification of the gas turbine is as tabulated in Table 4.2 at ISO conditions (T=15^0C and ∅=60%).

Table 4.2: Data description of the gas turbine

Description	Values
Turbine inlet temperature	1112^0C
Air flow rate	139.96 Kg/s
Isentropic efficiency of compressor	85.4[%]
Isentropic efficiency of turbine	86[%]
Combustion efficiency	98.0[%]
inlet pressures loss	100 [mmH_2O]

Exhaust pressure loss	200 [mmH_2O]
Combustion chamber pressure loss	1.2 [%]
Fuel, LHV	Natural gas; 48230 [KJ/Kg]

Generally, the actual performance of a specific GT at different inlet air temperature depends on its design. Figure 4.6 illustrates the effects of ambient air temperature on the performance of the selected Acacia gas turbine considering 100% efficiency at ISO conditions. An increase in inlet air temperature from 15 to 38^0C on a hot summer day decreases the performance to about 81% of its rated capacity. In another words, every increase in inlet air temperature leads to about 0.8% deterioration of rated performance. Figure 4.6 shows also the heat rate increases for the same change in ambient air temperature by about of 10% of rated heat at ISO conditions. Increasing the heat rate means decrease in the fuel efficiency defined as fuel energy required per unit of electric energy generated.

Figure 4.7 presents the decrease in power without inlet cooling. With employing an inlet cooling method at different humidity conditions, the cooling provide higher output level when the relative humidity is 18% than at 60%. This is due to the ambient dryness that affects the gas turbines performance.

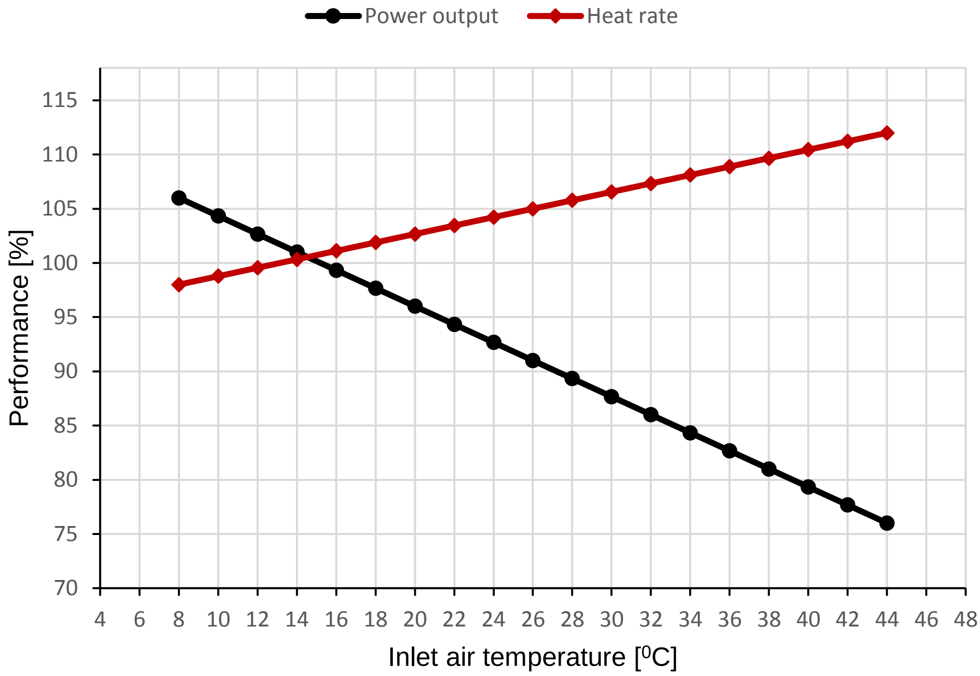


Figure 4.6: Inlet air temperature effect on Acacia gas turbine performance

This fact is related with the essence of the fogging cooling method. The ambient air passes the by the fog following a constant enthalpy-line (refer to Figure 5.3), but the resultant temperature drop is limited by the intake air initial.

When the fogging cooling technology is deployed Figure 4.8, the gas turbine thermal efficiency level is higher in relation with base-case (no cooling), as occurred for the power output results.

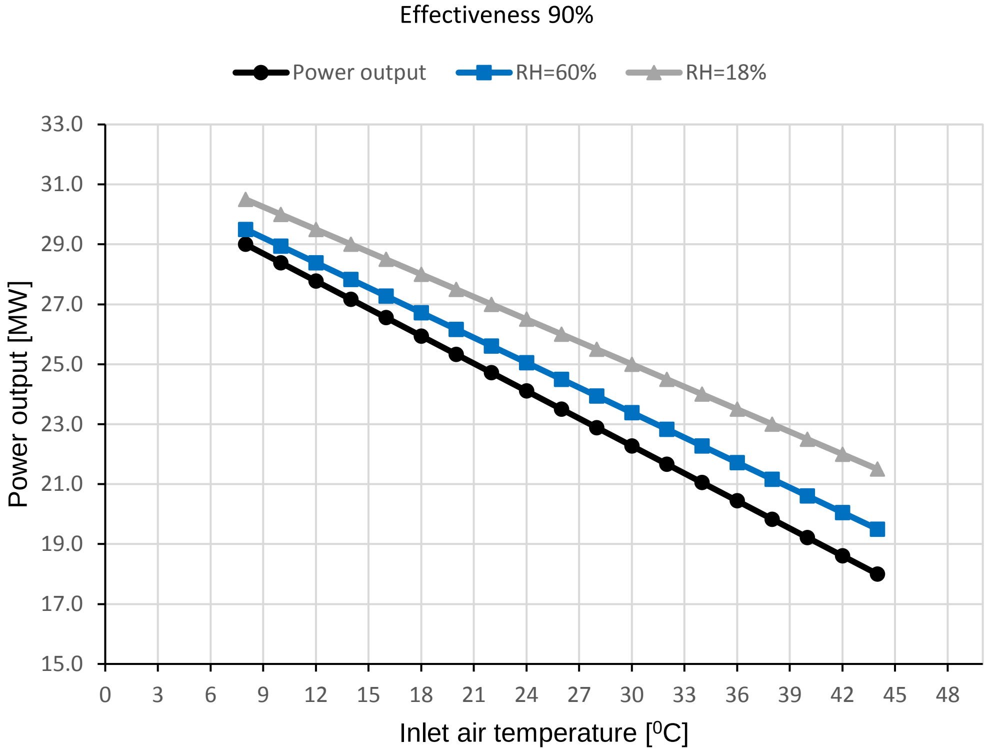


Figure 4.7: Inlet cooling effect on gas turbine power output at different relative humidity

From Figure 4.8 the gas turbine thermal efficiency sets in higher level in comparison with the thermal efficiency without cooling. The output power of TP4 Twin Pac is presented in Figure 4.9 for base-case condition. Note that the power output obtained is lower at base-case state when the intake air is not cooled.

Figure 4.10 presents the expected temperature decrease obtained using the fogging cooling method. Data were obtained with collaboration with AMCO. Different fogging cooling effectiveness values were simulated showing that a larger temperature decrease is achieved when the effectiveness is higher, as predicted.

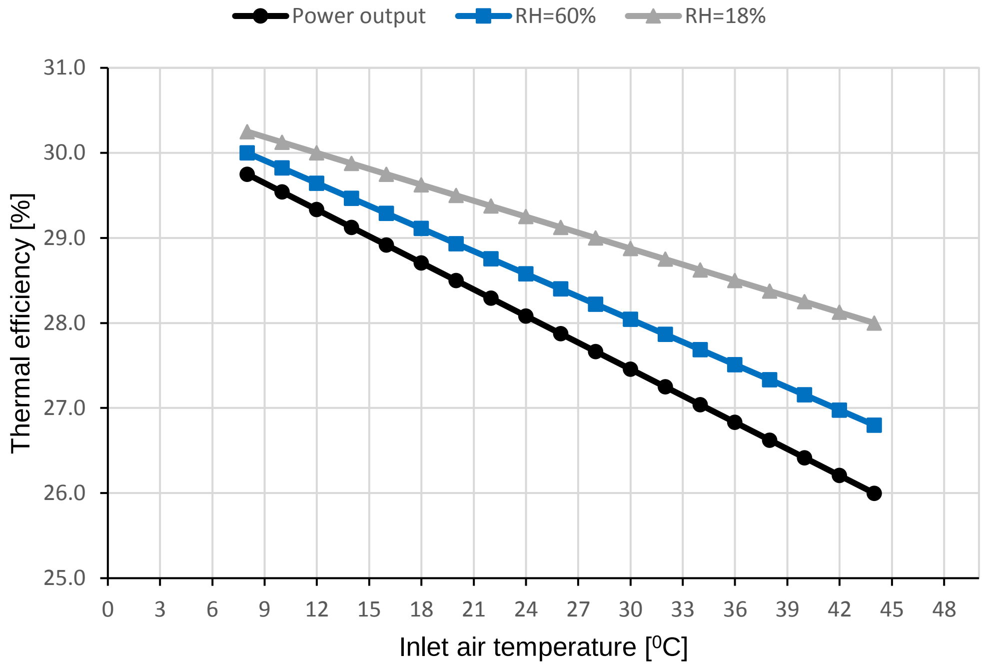


Figure 4.8: Evaporative effect on GT thermal efficiency at different relative humidity

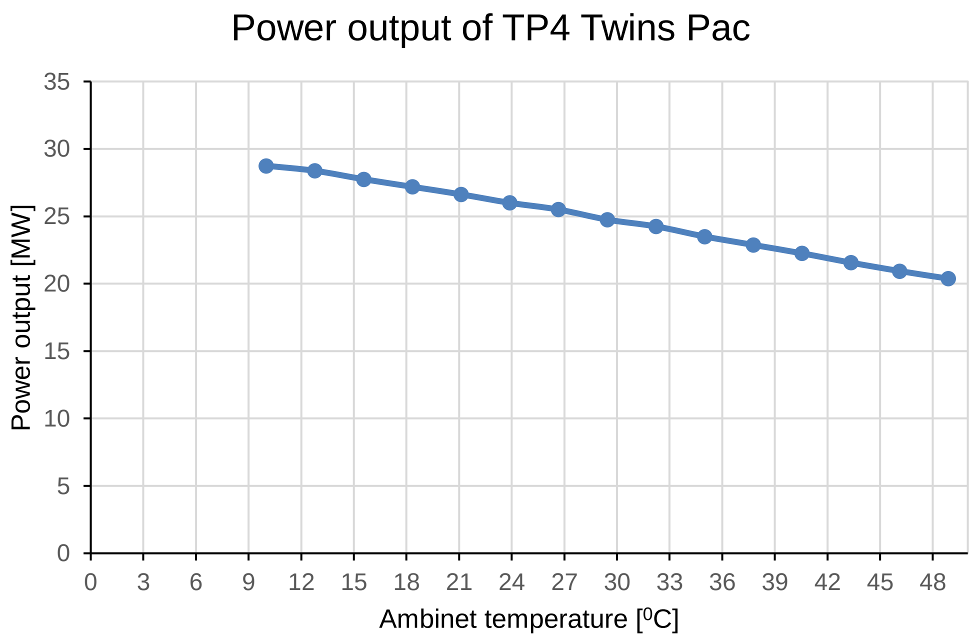


Figure 4.9: Base Load Power Output vs. Ambient Temperature

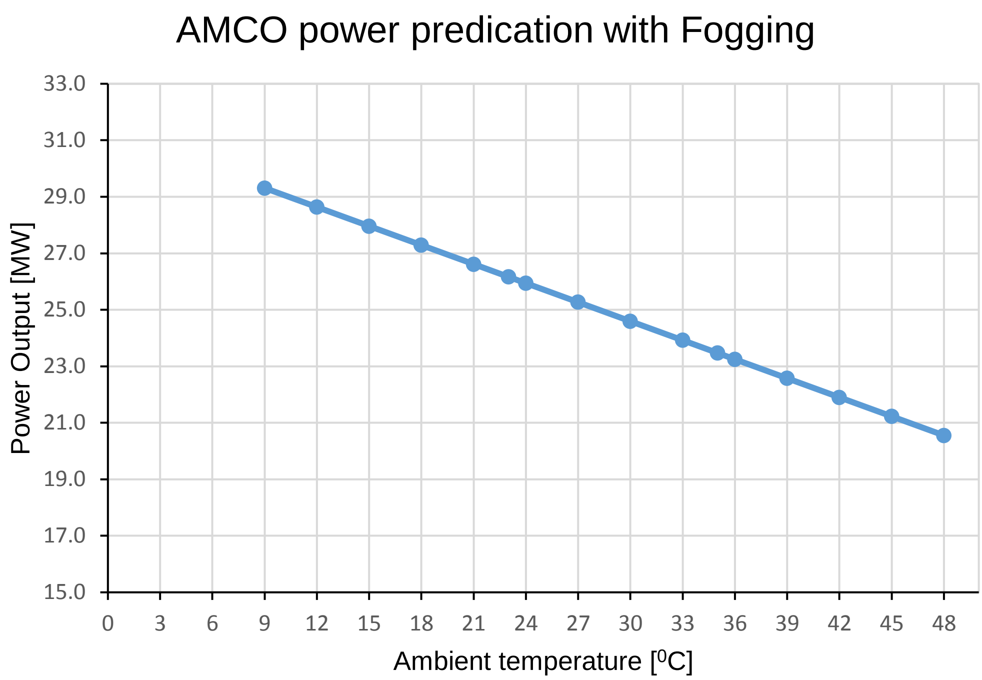


Figure 4.10: Simulated power output production with fogging

4.7 Summary

At the beginning of the chapter, a general review of the inlet cooling technologies and its classifications were presented. A comparison of different inlet air cooling technologies with key benefits and drawbacks were conducted. A discussion of economic and environmental perspective of inlet air cooling was also given. Lastly, analyses of Gas Turbine (GT) performance with ambient temperature and humidity are elaborated.

As previously mentioned, the focus of this study was to increase the power output of Acacia power station at hot temperatures in Summer. In this respect, a different of technologies were considered and the best suited option was the high pressure fogging system. The results indicate that the inlet air cooling technologies are primordial to ensure the gas turbine stability production in hot conditions.

It is vital to understand the additional components required of any cooling system will add additional cost of the power plant. However, these added components present an inferior cost when compared with a large simple-cycle gas turbine engine.

Therefore, the best cooling method must consider several factors as gas turbine capacity, load operation profile, site location, ambient temperature, desired cooling potential and economic feasibly. In this context the chapter considered the different types of inlet air cooling technologies. The next chapter considers a system description of the high pressure fogging technology as the best choice for this project.

CHAPTER FIVE

HIGH PRESSURE FOGGING SYSTEM

5 HIGH PRESSURE FOGGING SYSTEM

5.1 Introduction

The high pressure fogging system has been employed in the gas turbine industry since the late 1980s (Jones, C. and Jacobs III 2000). This technique applies a direct evaporative cooling which consists of small size particles, ranging from 5 to 20 microns, injected into the turbine intake and thereby providing cooling (Meher-Homji & Mee III 1999). According to Turbine Inlet Cooling Association technology overview-(TICA) since water is directly evaporated in the intake air stream, it has to be clear of any mineral salts, and other impurities (Bastianen & Escue 2009). Hence, the water used in fogging systems is generally demineralized. This can be achieved by reverse osmosis technology. The main difference between traditional evaporative cooling and fogging is that the latter has shown a high percent of effectiveness and it can reach wet bulb temperature even in high humid conditions.

The main components of a typical fog system are a series of high-pressure pumps that are fixed on a skid, PLC with temperature and humidity sensors, and an array of fog nozzles mounted in the inlet air duct.

Fog generation requires high-pressure demineralized water between 70 bar to 200 bar to a special designed a set of nozzles array. A study was done by Chaker *et al*. has shown the effects of different type of nozzles (Chaker et al. 2002b).

5.2 Fog system components

5.2.1 Pumps

The typical pumps used to generate 70 to 200 bar pressures used for gas turbine inlet air fogging systems are the positive displacement ceramic-plunger stainless pumps with stainless steel heads (Meher-Homji & Mee III 1999).

The location of fog skid should be as close as possible to the final distribution manifold. A typical skid outline is shown in the Figure 5.1.

Figure 5.1: MEE fog high pressure fogging system (Envitech 2014)

5.2.2 Control unit:

The control unit is an important component of a fogging system which consists of a PLC where it is mounted on the high-pressure pump skid along with temperature and humidity sensors to measure the air ambient and humidity. It controls the amount of evaporative cooling that is needed for a particular case. The system can operate on and off automatically to match the ability of the ambient air to absorb water vapour. In addition, the control system monitors pump skid operating parameters such as water flow rates and operating pressures and it indicates when emergency occurs.

5.2.3 Positioning of fog nozzles

There are two main methods of locating the inlet fogging system, either positioning them upstream or downstream of the filters as shown in Figure 5.2

5.2.3.1 Downstream of the inlet filters

This is the most common location for the fog nozzle. It is placed downstream of the air filters and up stream of the silencers and trash screen. In this way there is more residence time for the droplets to evaporate and it only requires minor modifications to the turbine inlet structure.

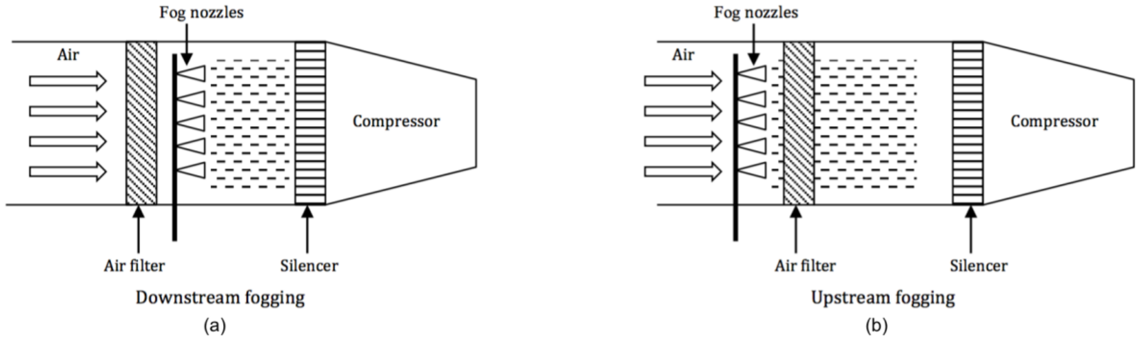


Figure 5.2: (a) downstream positioning, (b) Upstream positioning of inlet fogging

5.2.3.2 Upstream of the inlet filters

The advantage of locating the fog nozzle this way is that the installation can be achieved without outage time. However, a fog droplet filter must be added downstream of the fog manifold to remove any un-evaporated fog. The downside of this set-up is that it requires more fog nozzles, more water, and is generally more expensive to install and operate. This technique might be a cost-effective option for some operators who have experienced excessive loading of the inlet air filters, as fog scrubbing has been shown to increase air filter life (Meher-Homji & Mee III 1999).

5.3 Water quality consideration

As mentioned before, different types of gas turbine inlet cooling techniques are already available to cool the inlet air ambient temperature. These include direct evaporative techniques such as wetted media and the fogging system. Other methods include indirect evaporative cooling such as Mechanical refrigeration system. Each of these methods requires different water quality consideration. The water quality can play a considerable role as it can result on one hand additional operational costs to the turbine operator but on the other hand it eliminates blade corrosion. Since the study of this research focus is primarily on the direct method of fogging system, water quality consideration would be considered in this section.

According to a technical report by (Bastianen & Escue 2009) there are three factors associated with direct evaporative cooling techniques in gas turbine generation. These are listed as follows:

- water intake of the compressor section of the turbine;
- the quality of vapours exposed to the inlet air stream through the evaporation process, and the quality of the cooling water.

The supplier typically provides the specification of the water quality requirements. The specification may vary depending on the operational location and the availability of quality water. However, the general recommendations for water quality for wetted methods are recommended by one large supplier (Munters Engineering Bulletin [WTGT-0406]) and are as tabulated in Table 5.1.

Table 5.1: Typical water quality specification for GTIC (Bastianen & Escue 2009)

Calcium Hardness (as $CaCO_3$)	50-150 mg/l
Chlorides	<50 mg/l
Conductivity	50-750 micromhos/cm
Total Dissolved Solids (TDS)	30-500 mg/l

Suppliers do not recommend brackish water, seawater and reclaimed water. Special care must be exercised for the water quality used in fogging system as water is directly evaporated in the intake air stream. This is due to the primary concern of the possibility of blading erosion and the risk of small particles that can damage fog nozzles and possibly even compressor blading that occurs when using poor water. Therefore, special treatment is often required if it is to be used for gas turbine inlet air-cooling. Demineralization is one of the technical solutions for water treatments. It is a process to remove minerals that are dissolved in the water in iron form. Demineralized water can be produced by reverse osmosis technology, among other methods.

An alternative to direct evaporative method is indirect evaporative cooling. In this technique, however, the air passing into the turbine does not come in contact with the secondary cooling air stream. Therefore, it is not contaminating the inlet air with salts, minerals, or water vapour. Hence, water quality consideration is less stringent in comparison to direct evaporative methods. Table 5.2 provides general groundwater specification.

Table 5.2: Groundwater specification (Bastianen & Escue 2009)

Calcium Hardness (as $CaCO_3$)	1165 mg/l
Chlorides	2116mg/l
Conductivity	10420 micromhos/cm
Total Dissolved Solids (TDS)	7060 mg/l

* Seawater is typically at a level of 56000 micromhos/cm

5.3.1 Corrosion consideration

A particular concern was developed in the early years of adopting the application in gas turbine cooling with regards to compressor blade corrosion from the fog. Generating small droplets and the use of demineralized water eliminate the occurrence of this phenomenon. Some studies concerning the blade corrosion can be found in (Sexton, Urbach, and Knauss 1998).

Water quality considerations can be a major factor in gas turbine inlet cooling. Clean water is usually not available where gas turbine power plants are located. Water treatments are required when fog inter-cooling is desired, which, can be achieved by RO in order to mitigate inlet air contamination for the gas turbine blades. Indirect evaporative cooling offers a simple and less stringent water quality requirement compared to direct evaporative methods as mentioned earlier. However, it is essential that the precautions are taken to minimize the risk of any damage that may accrue by water leakage from the secondary airflow using poor quality water.

The importance of water quality can be overstated, particularly when fogging system is required.

5.4 Water requirements calculations

A general estimation method of calculating the amount of water required for a gas turbine inlet cooling can be illustrated in Figure 5.3 with airflow capacity of 200 kg/s. It is better to demonstrate an example at a temperature of 35^{0}C and dry bulb temperature of 43% RH on the psychometric chart (in this case point 1). The moisture content at this condition is 0.015 kg of H_2O/kg of dry air. Assume that the air is cooled to the ambient wet bulb condition (100% RH as the ending condition). Moving left up the constant wet bulb temperature line until saturation is achieved (point 2). The moisture content corresponding to this condition is 0.019 kg of H_2O/kg of dry air. Hence, the amount of moisture to be added to the air stream to achieve the wet bulb temperature (WBT) is 0.045 kg of H_2O/kg of dry air. Thus, the theoretical amount of water required to cool 200 kg/s air by 10.5^{0}C is 5.4 liters/min.

$$0.045kg \times \frac{200kg}{sec} \times 60\text{sec} = 540l/min$$

The reason for injecting water droplets is to increase the relative humidity of the airflow by evaporation of the droplets, which in turn leads to a decrease in the air temperature, resulting in an increase in the air mass flow due to increased air density.

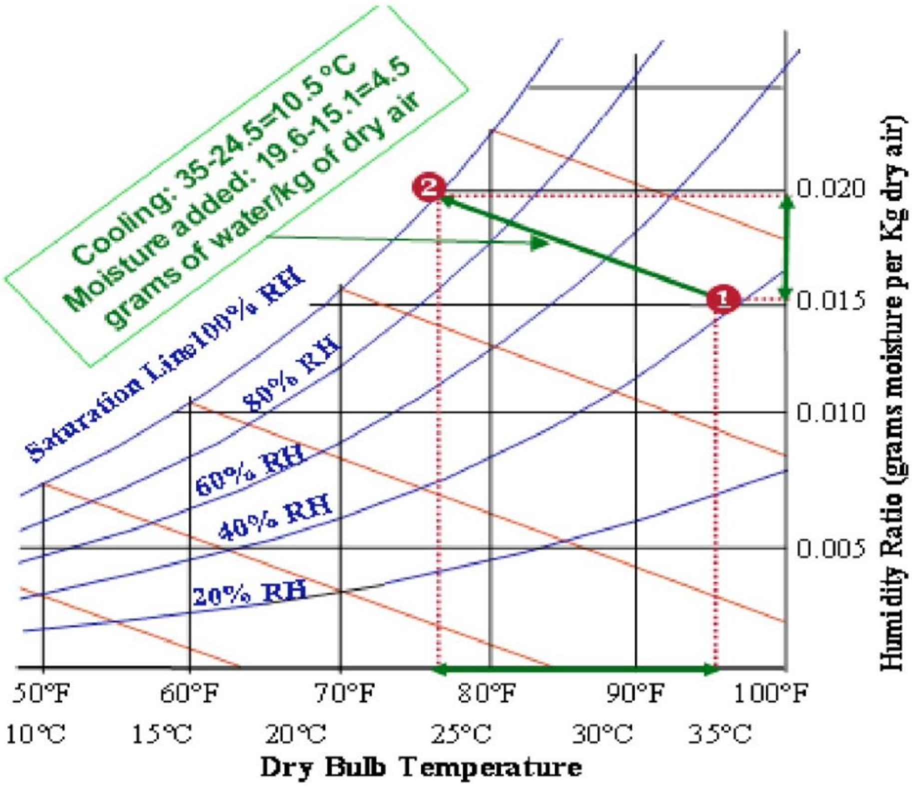


Figure 5.3: Psychometrics of inlet cooling (Bharbava *et al*. 2007a)

The evaporative cooling effectiveness is defined as

$$E = \frac{DBT - CIT}{DBT - WBT} \tag{5.1}$$

Where

DBT is dry-bulb temperature (^{0}C)

CIT is compressor inlet temperature (^{0}C)

WBT is wet-bulb temperature (^{0}C)

5.5 Gas turbine cooling load estimation

In this work, to estimate the water requirements for fogging cooling system, 98% wet-bulb temperature have been assumed (Kraneis 2000). Taking into account the effectiveness of the adiabatic process in fogging cooling systems with ε, the temperature of air exiting the cooler is then:

$$t_{a,exit} = t_{db} - \varepsilon \,.\; (t_{db} - t_{wb}) \tag{5.2}$$

Where $t_{a,exit}$ is the temperature of fogged water, t_{db} is the ambient temperature of air dry bulb and t_{wb} is the ambient temperature of wet bulb. Using the assumption that the evaporation is adiabatic and the temperature of the liquid water being fogged is equal to $t_{a,exit}$, the air water load $x_{a,exit}$ is

$$x_{a,exit} = \frac{c_{p,a}\left(t_{db} - t_{a,exit}\right) + x_{db}(c_{p,a}\,.\; t_{db} + r - c_{p,w}\,.\; t_{a,exit})}{c_{p,v}\,.\; t_{a,exit} + r - c_{p,w}\,.\; t_{a,exit}} \tag{5.3}$$

Where X is being water loading or specific humidity of exiting air from fogging coolers $c_{p,a}$ is specific heat at constant air pressure. Hence, the air condition at the exit of fogging cooling systems can be evaluated. The difference of temperatures between the air dry-bulb and the exit is then used to estimate the impact of the fogging technique on improving the power output of the gas turbine. The mass flow of liquid water $\dot{m}_w$, for the cooling can be given by

$$\dot{m}_w = \frac{\dot{V}_a}{v_a}\,.\left(x_{a,exit} - x_{db}\right) \tag{5.4}$$

Where $\dot{V}_a$ is being the volume flow rate of ambient air into the compressor (m^3/s), v_a is the specific volume of wetted air per kg of dry air (m^3/kg).

The variation between the behaviour of power output and efficiency of the gas turbine to ambient temperature can be designed by presenting the thermodynamic equations: where power input to the compressor

$$P_c = \dot{m}_{air}\,.(h_2 - h_1) \tag{5.5}$$

The considered gas turbine unit has the specifications listed in Table 5.3

Table 5.3: Gas turbine unit specifications

Power output at base load (MW)	28.5

Efficiency (%)	28.8
Heat rate (kJ/kWh)	11285
Inlet air mass flow rate at base load (kg/s)	144.24
Inlet air volume flow rate at base load (m^3/s)	119.75
Turbine inlet temperature at base load (^0C)	1104
Exhaust gas temperature (^0C)	532

The power capacity and economics of a GTIC system depends on various parameters, including the following (Andrepont *et al*. 2012):

i. gas turbines design and characteristics;

ii. heat rate versus compressor inlet air temperature

iii. power output versus inlet air temperature

iv. air flow into compressor

v. parasitic load water usage

5.6 High pressure fogging in industry

In recent years, high pressure fogging has been an addition to the technology list of inlet air-cooling where it has been installed on based-loaded and peaking gas turbine units. Cases show that operators tend to install the fog nozzle manifolds downstream of the air filters and before the silencers. Unlike the upstream positioning, it requires some modifications to the turbine inlet frame and therefore it requires one to two outage days. As previously mentioned, a fog system can also be installed upstream of air filters. Thus, it requires an elimination of fog droplets to prevent wetting of the air filters. However, no outage time is required. Fog system installers pay a special attention to the design inside the mounting of the nozzles manifolds. It must be designed appropriately to avoid small parts from breaking off inside the compressor

Increasing the ultimate generating capacity without increasing the size boundary is the goal for gas turbines future developments. In a description project, decision was made to further increase the power of Dickerson Station. The plant consists of three Combustion Turbine units with inlet air evaporative cooling to be installed. This allows additional power to be produced and increases overall efficiency. For the GE Frame 7F, a reduction of 13^0C in inlet air temperature would result in a 3% increase in power and an associated 1.2% in heat rate. Therefore, the result is a gain of power production, while lowering the associated emissions per MWhr generated (Mirant Cop. Access 2002).

In a case study reported by the Electric Power Research Institute (EPRI) in 1995, a General Electric MS7001 was used and a fogger evaporative cooling system injected 23 gpm upstream of the inlet air filters. The result was a 10% boost in power (from 61 MW to 67.1 MW) (Giampaolo 2003)

An advanced stage of high pressure fogging application known as fog intercooling has shown a further increase in power augmentation from the conventional inlet air-cooling fog system. The idea of this concept is to inject more fog into an air stream. The vapour fog droplets are carried into the compressor and evaporate when the air is heated by compression. According to a comparative guide to inlet air cooling (Mee Industries, Inc)

5.7 Fog design philosophy

The most important element in fogging design is supplying a pumping system of sufficient capacity to cool the inlet air in order to maximize the potential power capacity under most critical hot weather conditions when electric power demand is high and output is the lowest.

At the time of this work, we have not received confirmation regarding technical data from Acacia power plant for the specific TP4 Twin Pac gas turbine. Therefore, some necessary assumptions were taken in order to complete this study.

Generic data for a gas turbine were used in order to estimate fogging water requirements and forecast performance enhancement.

5.7.1 Design Conditions

All calculations were based on the following assumptions

i. place of gas turbine: Cape Town, RSA;

ii. critical design point: 35^0C and 66% relative humidity (29.3^0C WB)

iii. GT generates approximately 28.5 MW at ISO

iv. air flow is 139.96 Kg/sec at ISO conditions.

As mentioned before, the purpose of fogging system is to cool the inlet air during periods of hot weather, thereby permitting the GT to operate more efficiently and increased power output.

The saturation fogging systems are sized to bring the inlet air to within 2.2^0C of the wet bulb temperature- this saturation approach is best applied in hot, dry climates and in installations where the operator does not wish to utilize overspray.

Using demineralized water, the inlet cooling system cools the gas turbine's inlet air, increasing the power generated and decreasing NO_X and CO per unit MW and, decreasing heat rate.

5.7.2 Overspray

Defined as injecting an amount of atomized demineralized water and above the quantity of water required to achieve 100% air saturation, this amount of overspray water droplets is evaporated under the heat of compression, thereby increasing the mass flow which increases power output beyond what is achieved by saturation only.

5.8 Scope of supply

The cooling system shall be designed to fit within the inlet filter house as shown in Figure 5.4 and Figure 5.5. The scope of supply includes

i. inlet cooling water spray delivery system- nozzle array,

ii. water transfer system- pump skid

iii. 15 m water piping from pump skid to spray delivery system

iv. controls and instrumentation.

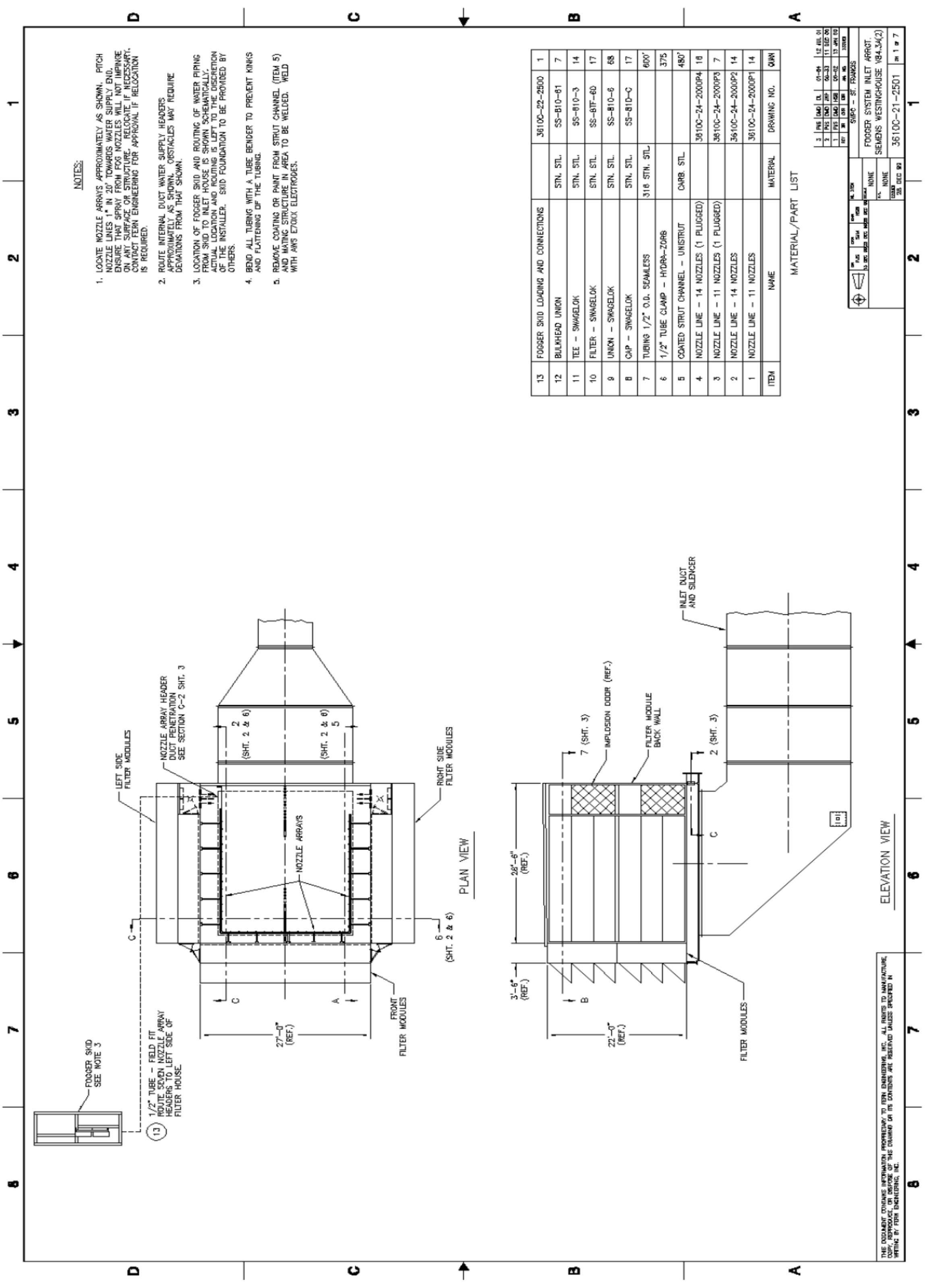


Figure 5.4: Designed layout fogging system-A

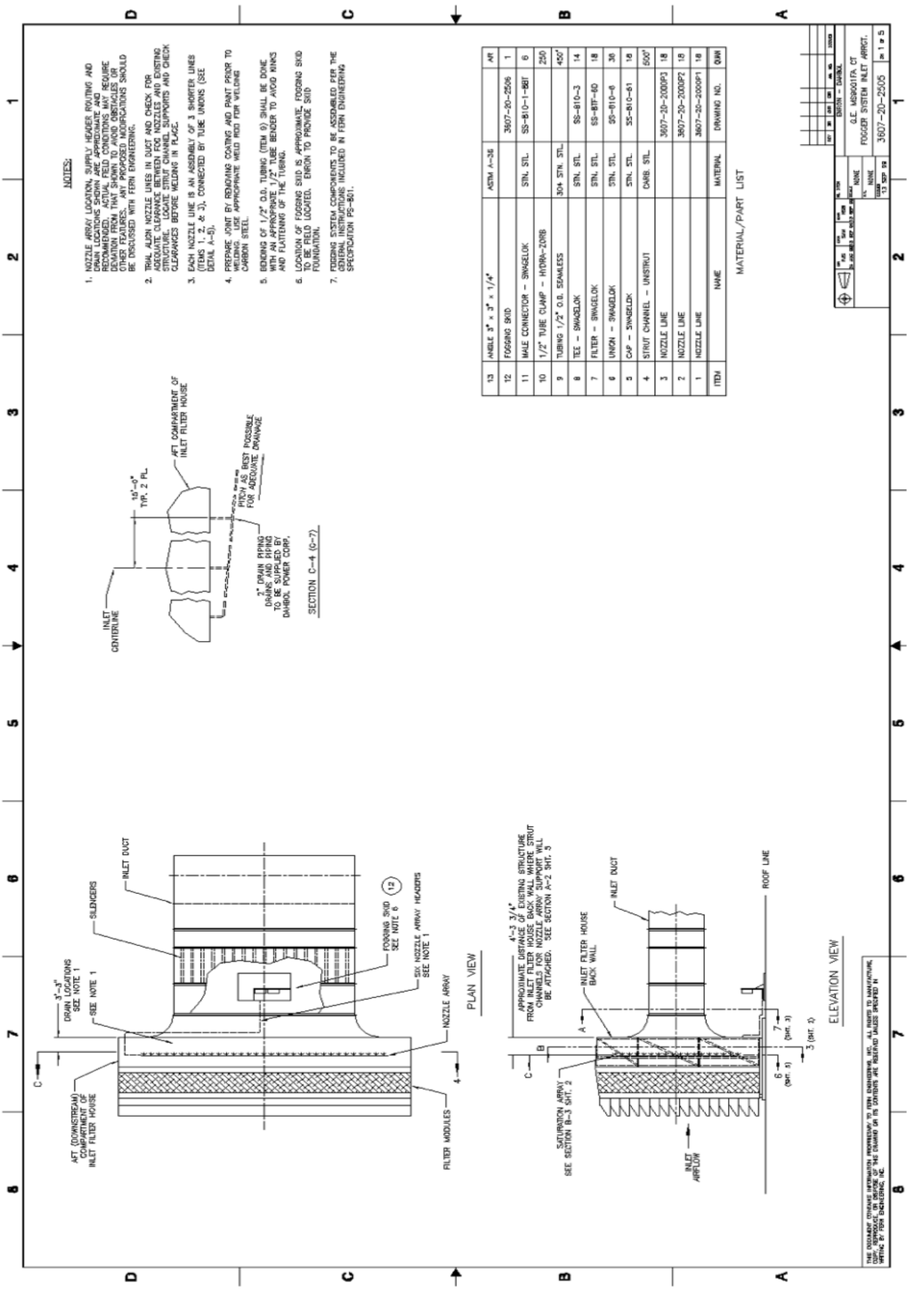


Figure 5.5: Designed layout fogging system-B

5.8.1 Design conditions for GT

The system is to be designed for the following conditions as shown in Table 5.4:

Inlet parameters

i. Compressor inlet air flow: 139.96 kg/s at ISO

ii. Elevation: 250 m estimated

Table 5.4: Fogging system designed conditions

Dry Bulb	T_{amb}=35^{0}C
Wet Bulb	T_{wb}= 29.3^{0}C
Rel. Humidity	66%
Target C.I.T.	31.5^{0}C
ΔT	5.7^{0}C cooling

5.9 Performance estimates of GT

Based on the performance curve supplied, cooling to saturation the inlet temperature by 5.7^{0}C should provide a 5.9% increase in power output. On other words, a 1.68 MW increase in power

i. If a 1% overspray is added, the turbine will gain a total of 11.4% power output above no-cool conditions

ii. If a 2% overspray is added, the turbine will gain a total of 16.9% power output above no-cool conditions.

To demonstrate the performance of TP4 Twin_Pac, Table 5.5 is included contains all the cooling parameters.

Table 5.5: TP4 Twin Pac Performance estimate

Parameters	(1%/2%) Overspray	Fogging	No Fogging
GT Power (MW)	52.36/54.94	49.75	47
GT Inlet Temp (^{0}C)	29.3	29.3	35
GT Inlet Air Flow (Kg/s)	-	128.30	-
Fog Water (Lt/s)	-	21.7	0
Overspray (%)	1%/2%	-	-
Overspray (Lt/min)	85.2/170.4	-	-
Total Fog System Flow (Lt/min)	106.9/192.1	20.4	0
Ambient Wet Bulb (^{0}C)	29.3	29.3	29.3
Relative Humidity (%)	66.0%	66.0%	66.0%

5.10 Inlet fogging cooling system description

The inlet cooling system consists of the following

i. spray delivery system (see Figure 5.4 and Figure 5.5);

ii. water transfer skid;

iii. Piping between the water transfer skid and spray delivery system; and

iv. Control system and software, as required.

5.10.1 Spray delivery system

The spray delivery system includes the following

i. nozzle arrays consisting of 316L stainless steel tubes and nozzles;

ii. all mounting and support hardware; and

iii. stainless steel piping from the water transfer skids to the nozzle array (15m maximum supply).

The scope of supply includes drawings and specifications of the spray delivery system including, at a minimum, the following:

i. spray delivery system assembly drawing;

ii. spray nozzles installation drawing;

iii. tube array and nozzle installation drawing, and

iv. array support drawing.

The inlet cooling array consists of stainless steel tubes with spray nozzles. The nozzles are threaded into fittings that are welded to the stainless steel tubes.

The nozzle arrangement and number of nozzles selected are based on the required flow in order to introduce a uniform spray into the air stream to approach wet bulb temperature for the range of ambient conditions. The nozzle spacing and spray pattern are selected to minimize the potential for droplet agglomeration, as well as to minimize the potential for impingement on the duct walls and other in-duct components. Specific requirements for the nozzles are provided in a later section.

5.10.2 Inlet cooling array design arrangement

The inlet cooling array will be located in the inlet filter house downstream of the filters and will be designed to cool the air to within 2.2^0C of the ambient wet bulb temperature. Figure 5.6 shows a manifold consisting of over 600 fogging nozzles in a 80 MW class heavy-duty industrial gas turbine.

The inlet cooling array location will be selected to permit maximum residence time for the water droplets to evaporate. Nozzle spacing is selected so as to minimize droplet agglomeration caused by interaction of adjacent sprays.

The inlet cooling fogging array is designed such that the water flow can be supplied in increments of approximately 8.33 lpm, with 2.2°C of cooling control for the TP4 Twin Pac Gas Turbine. This feature provides sufficient flexibility to achieve near 100% saturation for the range of ambient conditions for the installation site, with minimal water consumption during periods with low potential cooling.

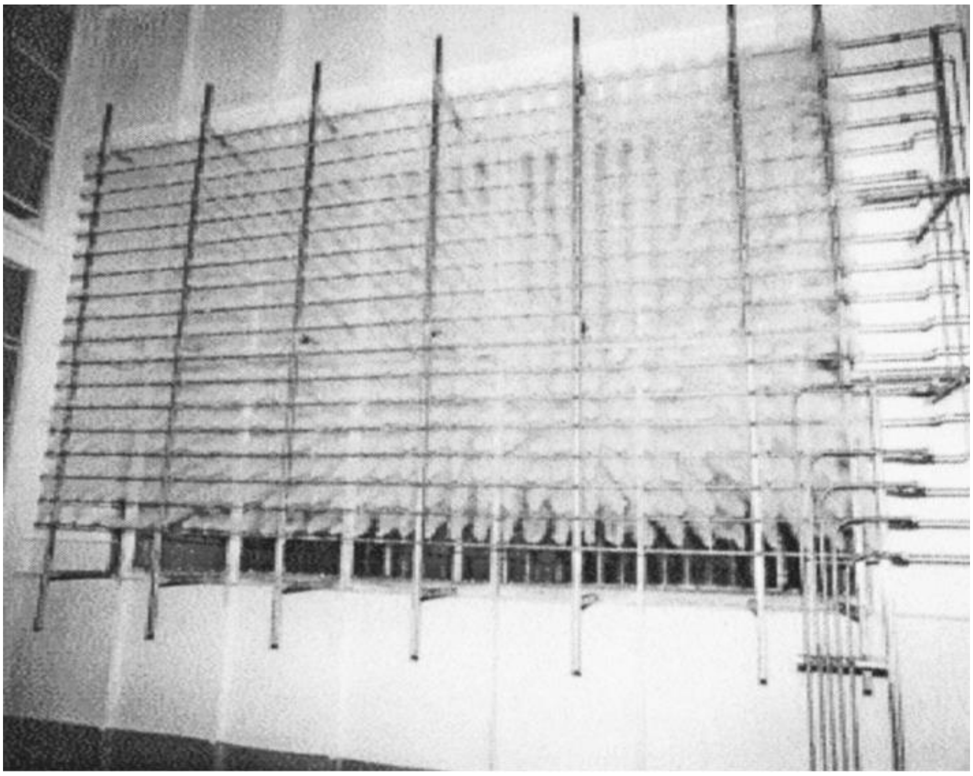


Figure 5.6: High pressure array (Chaker *et al.* 2003).

Components to monitor and control the required water flow and pressure to the spray nozzles will be mounted on the water transfer skid. Water pressure will be controlled to ensure proper droplet size distribution. Based on research completed for the *U.S. Electric Power Research Institute* regarding droplet size against water pressure, AMCO selected 207-bar (3,000 psi) for the operating pressure.

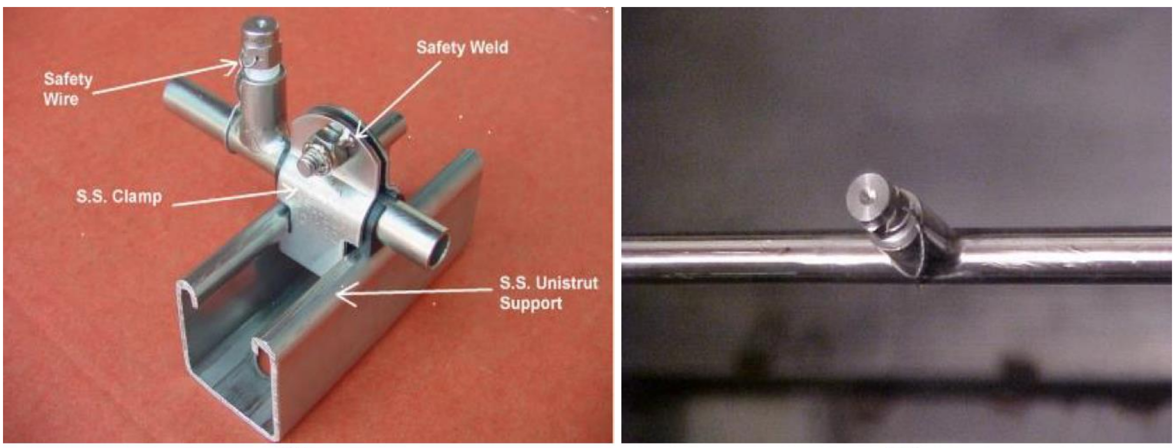
The required flow rate during operation will vary depending on the ambient design conditions. The flow will be controlled based on measured inlet air psychrometric properties and airflow to achieve near 100% saturation. Several options are discussed in later sections regarding the methods of software used to control the cooling systems.

Existing drains in the bottom of the compressor inlet plenum will drain excess water condensing on the walls. Additional drains and drain piping may be needed in the floor of the filter house and air duct to collect and drain any water that may accumulate on the floor. Specification of the additional drains were not considered in this work.

The arrays were designed so that individual tubes could be disconnected and removed from the inlet housing during periodic cleaning, testing or, maintenance inspections from within the filter house. The spray delivery system is designed to be fully drainable by gravity and configured for air-assisted blowout at the header end.

5.10.3 Nozzle array mechanical integrity

The nozzle tube array will be mounted to stainless steel supports that will be welded to the filter-housing duct. The tubes will be connected to headers. All duct penetrations of the headers will be sealed to eliminate outside air leakage into the housing. Figure 5.7 below illustrates the nozzle tube and the stainless support.



(A) (B)

Figure 5.7: Nozzle mechanical integrity (A): Nozzle clamp, (B): Nozzle tube

The overall support of the arrays will be designed such that induced flow and other undesirable vibrations are negligible. To insure against potentially damaging flow induced vibration, the tube and support design should be such that the vortex shedding frequency caused by the tube's interaction with the airflow is sufficiently different from the tube's natural frequencies.

5.11 Inlet cooling array zone flow

The total flow required for inlet cooling is calculated. The flow calculations utilize psychometric data, turbine airflow, dry and wet bulb temperatures, and droplet size distribution. The droplet size distribution used in the calculations was based on actual test data from the manufacturer nozzle at the appropriate pressure. The total flow was then used to predict the inlet cooling, and is plotted as a function of time. Another key factor is the length of time droplets remain as liquid in the air-stream before reaching the compressor inlet (referred to as droplet residence time). Calculations typically use a droplet residence time of 1.5 seconds. The analysis indicates that 22 lpm will drop the temperature from 35°C to 29.3°C. The nozzle tubes in the saturation array were configured in two zones such that spray can be supplied in percentages of total flow capacity. The number of nozzles, flow capacity and cooling for each zone are shown in Table 5.6 for the selected design point.

Table 5.6: Zone flow at design point

Pump	Zone	Nozzles	%Flow	Flow(l/min)	Cooling ^{0}C
1	1	32	39	8.48	2.22
1	2	50	61	13.25	3.48
Total				21.73	5.7

T_{amb}=35^{0}C, T_{wb}=29.3^{0}C, 5.7^{0}C cooling

The inlet cooling flow can be controlled in increments of 8.48 lpm or 2.22^{0}C. Each flow step is called a spray stage; thus the saturation array will have 2 stages of flow. It must be noted that only saturation fogging is controlled: overspray is manually controlled by the plant operators.

Flow control capability is accomplished through automatic or manual mode that turns the specific combinations of the 2 zones ON or OFF to achieve the required inlet cooling for the given ambient conditions. The system controls the flow to approach set point over a wide range of ambient conditions. The control system is described in later sections.

The 1% overspray is accomplished with 1 additional 85 lpm pump. This adds the additional 1% overspray for wet compression. Although we present overspray as only one (1) zone, the pumps are actually started with a 10-second delay from each other. The overspray mode is a manual procedure and is only ON or OFF. It can be activated from the skid or wired into the Digital Control System (DCS) to be activated from the control room. The 2% overspray is accomplished with 2 additional 85 lpm pumps. This adds the additional 2% overspray for wet compression.

5.12 Operational power requirements

The total power consumption based on maximum capacity with all pumps running is: 115 HP max i.e. 86 KW motor load plus 1.5kVA transformer load for instruments and control. This power demand will be supplied by solar PV system described in Chapter 6.

5.13 Demineralized water quality

It is customary to apply the same water quality standards, as defined by the turbine manufacturer, for water injection and steam injection processes. We recommend this practice for all fogging applications.

These quality standards are higher than those required by the fogging pumps, nozzles, or balance of the fogging system components. While the turbine manufacturer must define these guidelines based on specific engine requirements, these typical values often require less than 0.2 μSiemens/cm cation conductivity, Sodium less than 10 ppb and silica less than 10 ppb.

5.14 Nozzle design requirements

Laboratory tests confirmed that a single nozzle with water pressure at 207-bar produces a droplet size distribution with at least 90% of the mass being composed of droplets less than 19.2 μm in diameter. The mass median droplet size generated by a single nozzle is 12.5 μm in diameter. Nozzle density significantly influences the actual droplet size generated by a complete system of nozzles. The nozzles produced a droplet size distribution where 90% of the mass is composed of droplets less than 19.2 μm in diameter, optimizing evaporative efficiency. This criterion was used as a guideline to insure that the nozzles achieve the fast evaporation needed to reach saturation in the relatively short available residence times. Swirl Jet type nozzle has been tested at various pressures and its performance meets this criterion, as shown in Figure 5.8.

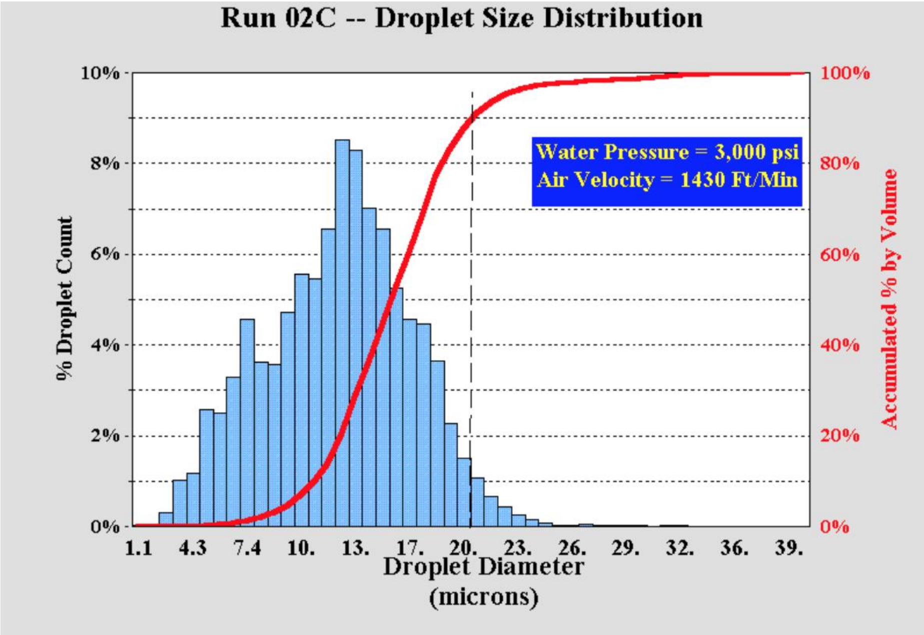


Figure 5.8: Droplet size distribution

The skid-mounted water transfer system interfaces with the demineralized water supply and with the piping to the nozzle array supply headers. The skid includes all components, valves, meters and instrumentation required to control and deliver the flow of water to the nozzle arrays. Figure 5.9 shows the designed Piping and instrumentation arrangement.

The saturation skid will include one constant speed positive displacement pump, electric motor, motorized valves, weather station, electrical service enclosures, water filter (if required), and instrumentation to monitor and control the total flow delivered to the nozzle array headers. This will handle saturation requirements.

The overspray skid includes two constant speed positive displacement pumps, electric motors, electrical service enclosures, water filter (if required), and instrumentation to monitor and control the total flow delivered to the nozzle array headers. This will handle the 2% overspray requirements for the system at the above conditions.

The skid-mounted water transfer system in Figure 5.9 includes the following

i. steel skid;

ii. high-pressure pumps;

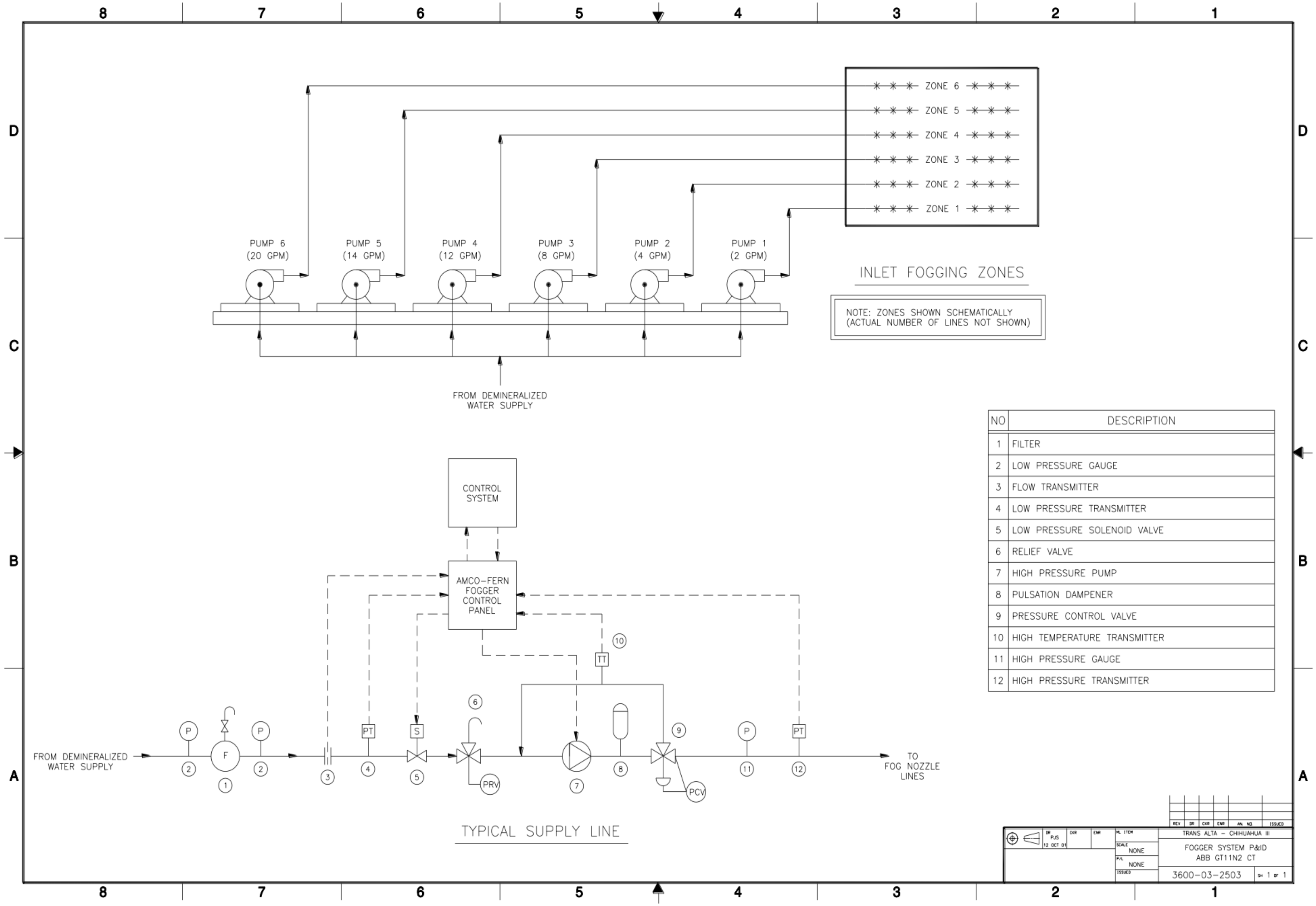
iii. control panel Inlet filter;

iv. pressure transmitters;

v. water and electricity connections to interface with customer-supplied water and electrical feeds; and

vi. required tubing, fittings and valves

The skid assembly would be based on the arrangement provided below in Figure 5.10. This includes a 22 lpm pump/motor system. The drawing of Figure 5.9 presents the designed skid arrangement. It has been assumed that the skid will be located so that no more than 15 metres of piping is required from the skid to the filter house and inlet duct.



INLET FOGGING ZONES
ZONE 6
ZONE 5
ZONE 4
ZONE 3
ZONE 2
ZONE 1
NOTE: ZONES SHOWN SCHEMATICALLY (ACTUAL NUMBER OF LINES NOT SHOWN)
PUMP 6 (20 GPM)
PUMP 5 (14 GPM)
PUMP 4 (12 GPM)
PUMP 3 (8 GPM)
PUMP 2 (4 GPM)
PUMP 1 (2 GPM)
FROM DEMINERALIZED WATER SUPPLY
CONTROL SYSTEM
AMCO-FERN FOGGER CONTROL PANEL
FROM DEMINERALIZED WATER SUPPLY
TO FOG NOZZLE LINES
TYPICAL SUPPLY LINE
NO
DESCRIPTION
1 FILTER
2 LOW PRESSURE GAUGE
3 FLOW TRANSMITTER
4 LOW PRESSURE TRANSMITTER
5 LOW PRESSURE SOLENOID VALVE
6 RELIEF VALVE
7 HIGH PRESSURE PUMP
8 PULSATION DAMPENER
9 PRESSURE CONTROL VALVE
10 HIGH TEMPERATURE TRANSMITTER
11 HIGH PRESSURE GAUGE
12 HIGH PRESSURE TRANSMITTER
TRANS ALTA - CHIHUAHUA III
FOGGER SYSTEM P&ID
ABB GT11N2 CT
3600-03-2503
SH 1 OF 1
SCALE NONE
P/L NONE
PJS
12 OCT 01

Figure 5.9: Designed piping and instrumentation arrangement

5.15 Cooling system controls

Control of the cooling systems can be accomplished with one of three methods; using a personal computer, using a PLC, or by direct programming into the plant's digital control system. A PLC-based control system was assumed and described herein. An Ethernet connection is supplied as standard equipment.

This section provides an overview of a PLC-based, automatically operated skid control system. The control system acquires data from the weather station instrumentation, the water transfer system, and from the existing gas turbine instrumentation. The system requires a weather station to measure ambient inlet air temperature and corresponding wet bulb temperature.

The Power Plant control room must supply a 4-20mA signal to the inlet cooling system to monitor compressor inlet temperature. Output from any instrumentation presently used at the plant may be used, so long as it is agreed that such instrumentation is adequate to control the system.

The inlet cooling system is capable of delivering a variable amount of water to the nozzle arrays. The PLC will transmit signals to the cooling system's control panel in order to control operation of the pumps and motorized valves as required, to achieve the specified set points and ramping schedules. The water flow control function will be accomplished exclusively by controlling the number of active nozzle tubes. The algorithm will include the logic to activate the nozzle array tubes to meet the necessary water flow.

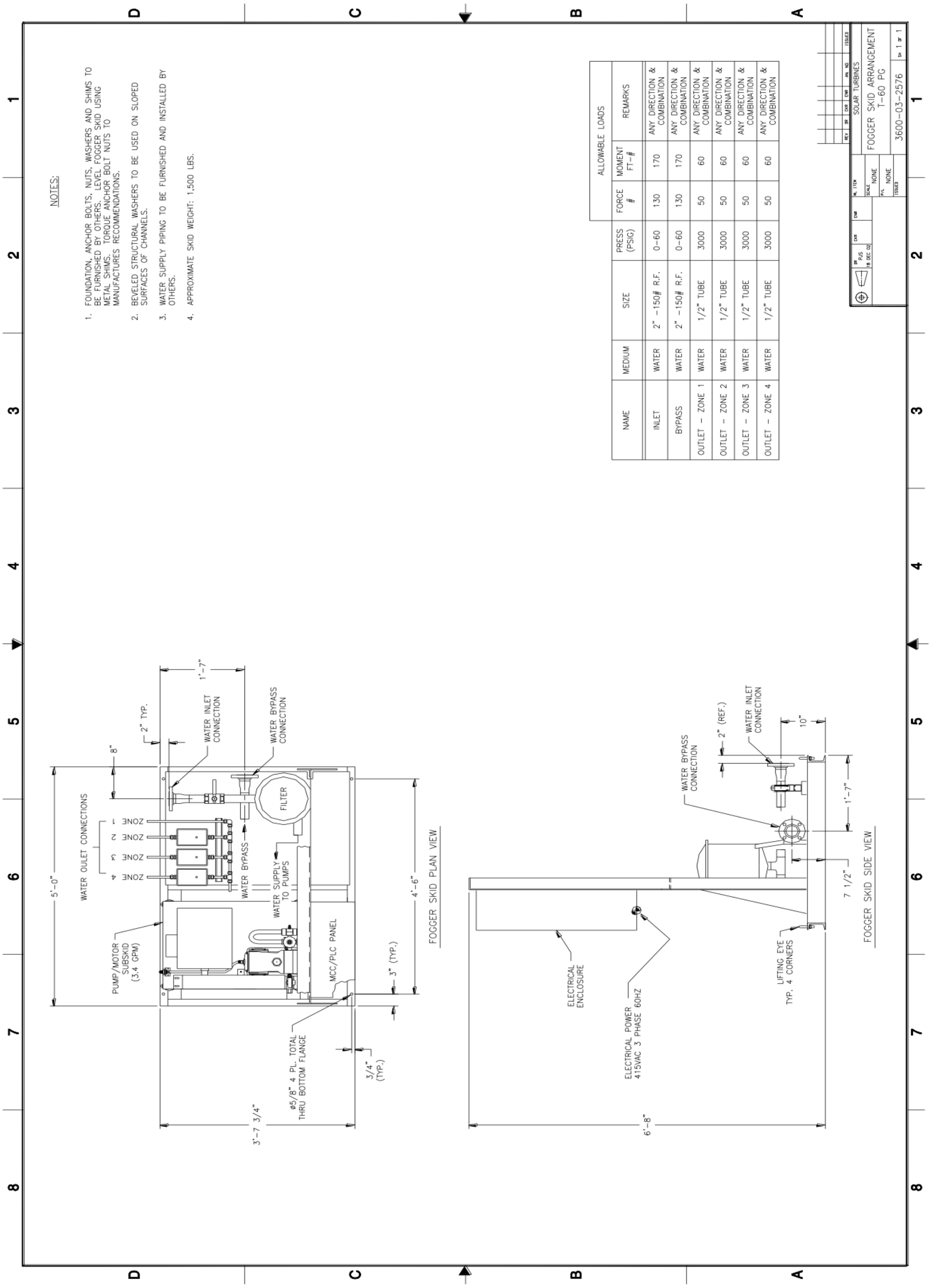


Figure 5.10: Inlet cooling skid arrangement and loading

The PLC control system will also receive signals from the system's transmitters mounted on the water transfer system skid to provide protection in the event of a malfunction. The software is designed to allow an operator to operate and monitor the system. The control system's diagnostics will monitor each function of the inlet cooling system and report on the type and location of the malfunction. The control system operator is provided with the following feedback data after start- up:

- motor operation status;
- alarms- low pressure for all pumps;
- compressor inlet temperature;
- ambient dry bulb temperature;
- Wet Bulb temperature, calculated; and
- predicted flow and stage number to achieve saturation ($T_{CI}=T_{WB}$).

5.16 Weather station instrumentation

Separate weather stations may be required for each skid and should be located as to provide representative ambient data representative for the gas turbines. This work assumes that the inlet temperature at the compressor inlet will be taken from sensors already on Acacia gas turbines. Data from this instrumentation will be used as feedback to monitor the cooling system's performance.

5.17 Stage flow operating conditions

An automatic control system automatically determines and set the required spray flow based on the current ambient conditions. The sequences of pumps and motorized valves in operation are pre-determined by the cooling system's control to provide the required flow.

The algorithm will produce the expected level of saturation predicted for each stage flow number. The controls will continually monitor the expected versus actual compressor inlet temperature and provide feedback data to adjust the flow as required to achieve the required level of saturation.

5.18 Advantages of Swirl Jet nozzles

Fogging systems for combustion turbines require precise designs. Therefore, the Swirl Jet nozzles are described below:

- better droplets distribution;
- doesn't have fragile external impingement pin;
- all nozzle components are 316L stainless steel;
- no higher wear items: Impingement pins have rubber O-rings;
- higher flow: (Swirl Jet = 0.26 l/min at 207 bar) while (Impingement pin = 0.21 l/min at 207 bar);
- less agglomeration; and
- full control of direction of fog.

Table 5.7: Swirl Jet diameter measurements

Nozzle type	Diameter (μm)
Swirl Jet	20
Impingement Pin	26

5.19 Summary

The most important concern to obtain an effective inlet fogging is to minimize the droplets size and to maximize residence time so that one can obtain complete evaporation. The higher the pressure, the smaller the droplet size. In this research project, 204 bar was selected in consultation with inlet fogging manufacturer known as AMCO. Swirl Jet nozzles have superior droplet distribution and consistent performance. Swirl Jet nozzles have a higher flow rate than impingement pins, which also means lower nozzle density and less chance for droplet recombination downstream. The path of distribution is less likely to lead to intersecting nozzle flows. Finally, the Swirl Jet nozzle is far more durable and damage-resistant with no delicate external impaction pin; no high wear items, and all 316L Stainless construction.

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CHAPTER SIX

SOLAR PV SYSTEM DESIGN

6 SOLAR PV SYSTEM DESIGN METHODOLOGY

6.1 Solar PV overview

Solar PV technology produces electrical energy by harvesting the energy produced by the sun. Hence, it can be applied to offset the conventional energy consumption of the cooling facility. PV gets its name from the process of converting light (Photons) to electricity (voltage), which is the PV effect. PV technology is clean energy reliable and easier to install for domestic and industrial energy. Such a system consists of an array of solar PV modules, an inverter, mounting structures, switchgear, and reticulation. Figure 6.1 provides a simple illustration of a typical solar PV system connected to the grid.

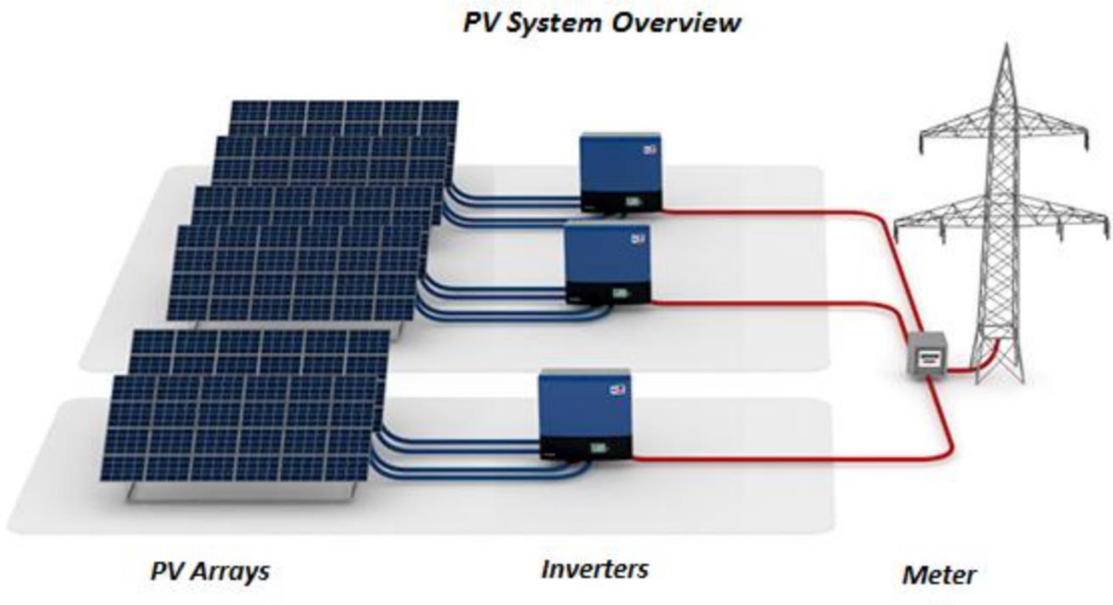

Figure 6.1: Schematic solar PV system overview

This chapter aims to give an insightful investigation of the proposed on-grid solar PV water system with its associated challenges and benefits to partially substitute the energy consumption of inlet cooling facility from the power plant. In addition, this work delivers a technical study to make an informed decision regarding the investment in solar PV system.

The evaluation of the project considers various factors include load profile, site preference, potential solar irradiation among others. Moreover, the aim is to present a preliminary design with layout structure to estimate the power yield per annum. The feasibility of the project is an assessment not just to provide clean power but also as an investment for the power plant. Cooling was considered to have a high electricity consumption profile. Most of the energy consumed during daytime is at a peak-on tariff that coincides with the potential of generation curve of a PV system. Therefore, such a system of this kind has been adopted in this dissertation.

The amount of energy produced by a PV power plant is subject to various factors, such as the type of PV module, the tilt and azimuth angle of the PV array, the ambient temperature, the solar irradiation, as well as the climate conditions for a particular location. This chapter also presents the modelling and simulation of a on-grid solar PV pumping system for inlet cooling fogging system. The model is consists of a PV array, DC/AC inverter and water pump.

In this chapter, the selection of a case study to determine the sizing components, and a performance analysis of the solar PV pumping. The system modelling was executed by computer program; the program software used was written by PSIM. The load requirements in terms of the water quantity and pressure were determined. Basically the load being the motor connected to the pump. The size of the water reservoir was also calculated by taking account of three cloudy days.

From the load profile of the fogging system, the proposed capacity of the solar PV system is considered to be $90KW_p$. Hence, the system is expected to generate an estimated 166000 KWh per year. This system is estimated to offset the requirement of the annual electricity consumption of the cooling facility from the utility grid.

6.2 PV water system components

A solar PV water pumping system typically consists of an array of modules depending on the size of the system, Variable Speed Driver (VSD), protection unit, and water pump.

6.2.1 PV modules

The PV array is the primary component of a PV water system that produces the energy, while the other components serve to transform and control the energy. Therefore, it is of the utmost importance to install reliable modules in order to ensure the success of the project.

In order to minimize the risk of a faulty system, it is important to invest in high quality modules. The modules should be manufactured according to high quality standards and should be accompanied by a linear performance warranty to ensure profitability and long-time steady performance.

It is recommended to choose well-known suppliers with a good track record and a local presence in order to ensure proper after-sales service and support. From own work experience in the field, SolarWorld modules are highly recommended for their robustness and long life performance.

6.2.2 Variable Speed driver (VSD)

The PV array produces direct current power at a relatively high voltage. The purpose of the VSD is to convert the power to alternating current at rated voltage to run and control the pump accordingly in terms of the available sun power. There are a lot of suppliers for variable speed driver and it is important that the VSD complies with the relevant safety measures and regulation.

6.2.3 Mounting structure

From an economic prospective, a fixed tilt mounted structure would be the most feasible option. The structure must be rigid and strong enough to hold the weight of the panels through different circumstances.

6.2.4 Balance of system

The balance of system includes switchgear, conductors and safety equipment. One important point for voltage drop is choosing the right cable sizes.

6.3 Load profile analysis

The proposed solar PV water pumping system should meet the water requirements to supply water demand for the inlet fogging cooling system. In order to secure enough water resource for the fogging system, a reservoir tank with capacity of $60m^3$ should be included in the system with all the pipe fittings. The water profile is based on the technical data sheet provided by one of the top leader in fogging industry by the name of MeeFog. Figure 6.2 is showing water consumption for 11^0C of inlet cooling and gas turbine power increases attainable provided by MeeFog Industries.

6.4 PV energy system with reverse osmosis integration

Desalination systems based on PV form is the largest renewable energy conversion method in potable water with RO being the most common technology. The good reliability and relative ease of installation are what makes solar PV system attractive. They include various systems components characteristics and seasonal variations of solar irradiance, wind speed and ambient temperature in the simulations.

6.4.1 Modeling and methodology

The overall system modelled is presented in Figure 6.3. The simulation was conducted in the MATLAB/Simulink environment. The aim is to establish relevant model parameters and operational characteristics performance of the system components. Figure 6.3 depicts the hardware components. For simulating the PV integrated system, it requires three basic energy aspects to be resolved in modelling:

6.4.1.1 Renewable energy conversion

Solar panels basically convert light energy into DC electricity. It should take into account PV panels current-voltage (I_V) characteristics when it comes into incorporating PV power into modelling.

WATER AND POWER REQUIREMENTS							
GAS TURBINE MODEL	ISO OUTPUT (kW)	kW 100˚F (38˚C)	WATER FOG FLOW		kW 80˚F (27˚C) SATURATION	POWER INCREASE (kW)	POWER IN-CREASE (%)
			GPM	LPM			
Alstom GT 8C	52600	41061	12.1	45.8	45980	4919	12.0
Alstom GT 11N	83880	70013	21.7	82.3	74920	4907	7.0
GE 5341N	24750	20252	19.0	71.0	22143	1891	9.0
GE 6541B	39615	32707	21.0	79.0	35500	2793	8.5
GE 7111EA	84920	69533	20.0	76.47	75033	5500	7.9
GE 7221FA	161650	128621	29.0	110.0	139998	11377	8.8
GE 9171E	126206	102777	28.1	106.0	111446	8669	8.4
GE LM2500+PK	27017	19001	5.5	20.8	22917	3916	20.6
GE LM6000PA	41020	25310	8.0	30.3	33475	8165	32.3
Solar Mars	10685	8443	2.8	10.6	9526	1083	12.8
W501 D5	109307	88153	25.0	95.0	95998	7845	9.0
SW501 F	171790	139596	30.1	114.0	150812	11216	8.0
SW V94.2	159410	133185	82.0	302.0	145237	12052	9.0
SW701 F	252560	206463	44.6	169.0	223512	17049	8.3

Figure 6.2: Typical water consumption of inlet cooling

- Solar irradiance: This parameter is the data input for the simulation. Global average irradiance data is available. For the purpose of this study, global irradiation models are used to predict hourly resolved over 365 days.

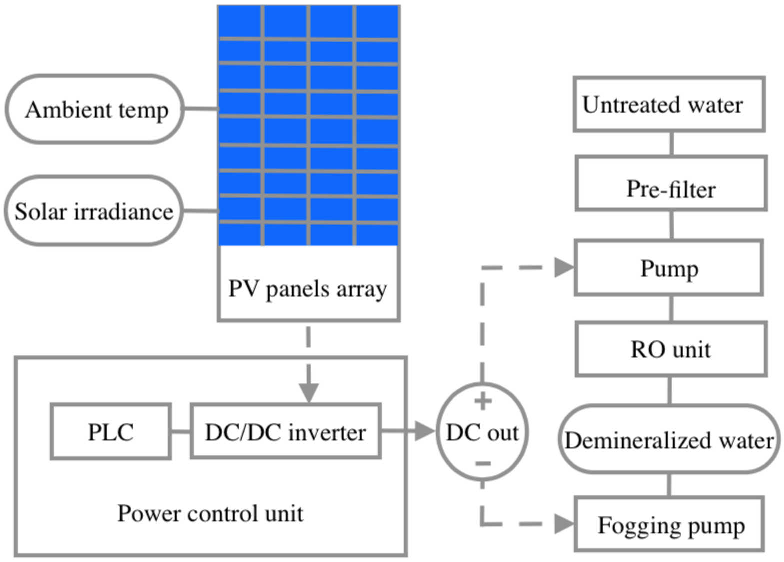

Figure 6.3: Layout of PVRO system components

- PV panels: panels manufactures provide basic hardware specifications and model parameters for PV panels. The (I_V) characteristics are shown at 25^{0}C and at different solar irradiance (w/m^2).

6.4.1.2 Control and power conversion

These components are responsible for converting DC power into AC power and is used to run the RO system. A system may also include means of storage represented as batteries banks and power controllers to regulate the voltage for charging/discharging the batteries as well as the operational status of the overall system at any time.

- Controller: The unit control consists of a PLC that has pre-set conditions of complicated methodologies and predictive techniques to regulate the operational status of particular energy system.

6.5 Yield assessments of the photovoltaic power plant

The selected site for the purpose of this work is Western Cape, South Africa with coordinates $33^0\ 52'\ 59.96''\ S, 18^0\ 32'\ 8.35''\ E$ at 23 m elevation above sea level with slope inclination 28^0 and slope azimuth -179^0 Northwest. The solar irradiation in-plane is annually about 2340 KWh/m^2 and annual air temperature at 2 m 17.3^0C. This amount adds to the feasibility of the project. Another advantage of available land for the system with ground mounting allows a proper design to face the solar north with an optimum title angle to maximize the yield performance. The proposed location is near to the main point from the grid at Acacia power plant which gives easy access to the inverter output to be connected to feed the facility at minimum power losses.

6.5.1 PV system information

In the table 6.1, the PV system information of Annual average electricity production: 166000KWh and Average performance ratio: 79.0% were provided.

Table 6.1: Solar PV system data

Provided inputs	
Location [Latitude/Longitude]	-33.884, 18.542
Horizon	Calculated
Database used	PVGIS-CMSAF
PV technology	Crystalline silicon
PV installed [kWp]	90
System loss [%]	14
Mounting system	Fixed mounting
Simulation outputs	
Slope angle [^{0}]	28 (optimal)
Azimuth angle [^{0}]	-179 (optimal)
Yearly PV energy production [kWh]	166000
Yearly in-plane irradiation [kWh/m^2]	2340

Year to year variability [kWh]	1790
Changes in output due to	
Angle of incidence [%]:	2.6
Spectral effects [%]:	0.6
Temperature and low irradiance [%]:	-6.2
Total loss [%]:	-21

6.6 Site assessment

This section provides an overview regarding the locality, solar resource, topography and electrical infrastructure associated with the proposed site. Figure 6.4 and Figure 6.5 show the geographic coordinates and the location of the proposed solar PV system location. The solar resource associated with the proposed site is adequate due to the favourable climate and solar irradiation, as illustrated in the solar radiation map of South Africa (Appendix B).

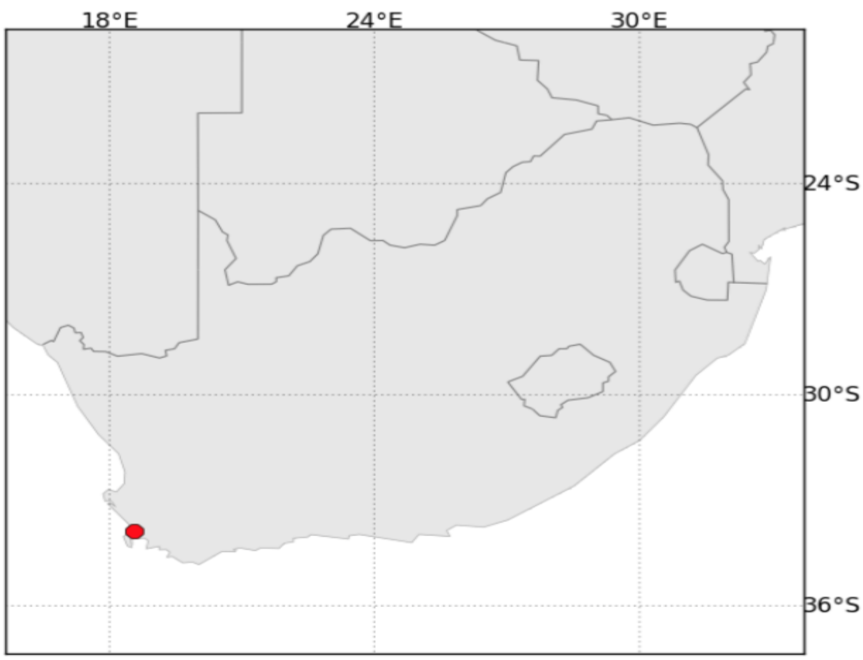

Figure 6.4: Geographic coordinates

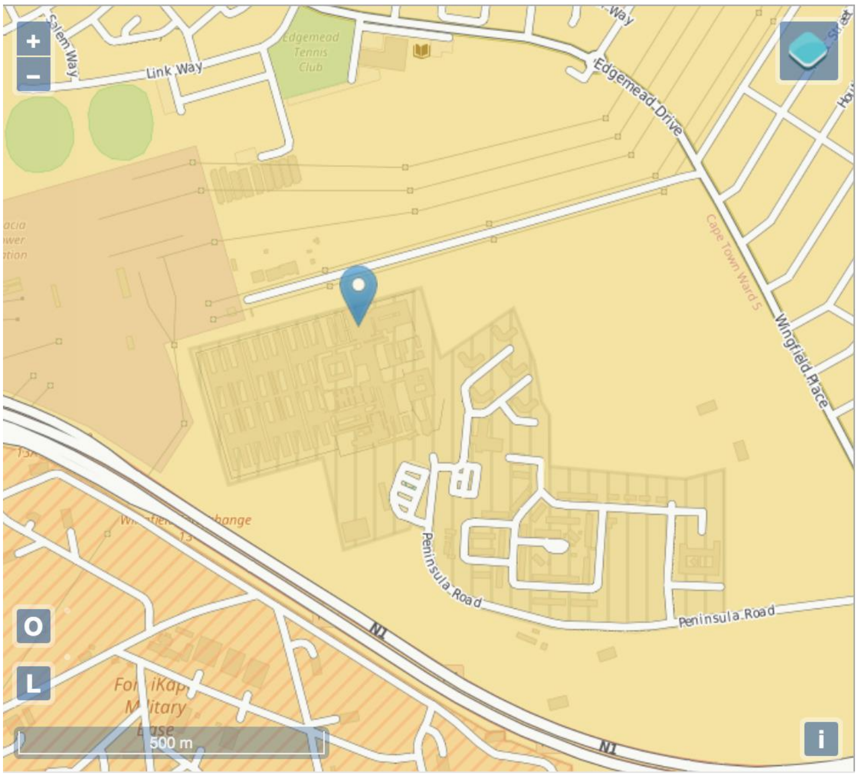

Figure 6.5: Geographic location for the solar PV system site (Google Maps)

6.7 Solar PV energy yield and solar irradiation

In order to forecast the annual energy production of the PV plant, a yield estimation has been conducted. The output of the system was estimated by applying the solar resource data provided by SolarGIS, a high-resolution climate database managed by GeoModel Solar. The amount of solar energy received on a surface at any given location depends on the altitude, latitude, time of day and year, local weather conditions and atmospheric effects, and orientation of the surface with respect to the sun. Figure 6.6 shows the average monthly energy production from the given system. While Table 6.2 presents the daily as well as monthly average energy production.

Table 6.2: Average daily and monthly energy production

Month	Ed	Em	Hd	Hm	SDm
1	555	17200	8.04	249	316

2	553	15500	7.99	224	526
3	511	15800	7.27	225	601
4	443	13300	6.2	186	630
5	338	10500	4.62	143	844
6	307	9200	4.11	123	422
7	335	10400	4.49	139	853
8	386	12000	5.25	163	710
9	448	13500	6.2	186	472
10	517	16000	7.28	226	421
11	537	16100	7.65	229	458
12	541	16800	7.83	243	339
Year	455	13900	6.4	195	149

Where

Ed Average daily energy production from the given system (kWh)

Em Average monthly energy production from the given system (kWh)

Hd Average daily sum of global irradiation per square meter received by the modules of the given system (kWh/m^2)

Hm Average monthly sum of global irradiation per square meter received by the modules of the given system (kWh/m^2)

SDm Standard deviation of the monthly energy production due to year-to-year variation (kWh)

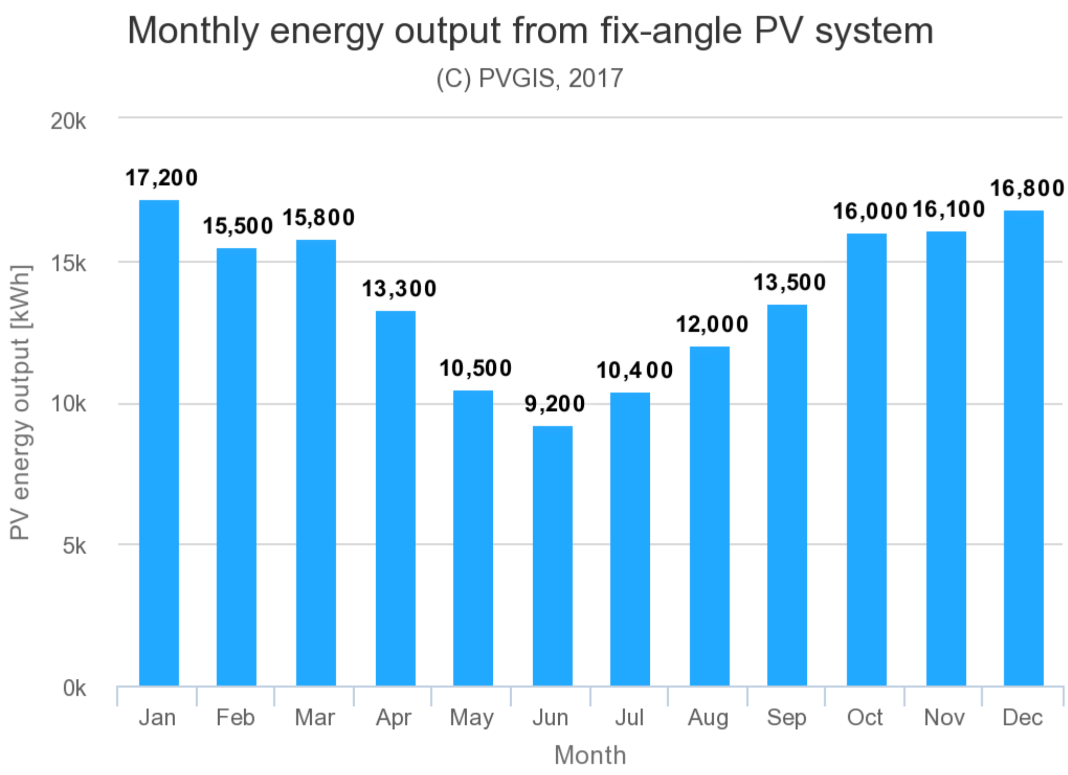

Figure 6.6: Average monthly energy production from the given system

The long term monthly average irradiation reaches 2180 KWh/m^2 per annum, resulting in a daily average of more than 5 KWh/m^2, which can be considered high average. The monthly in-plane irradiation is illustrated in Figure 6.7. Therefore, the proposed site is considered as a high priority site for the development of solar PV system in terms of the high levels of available solar resource.

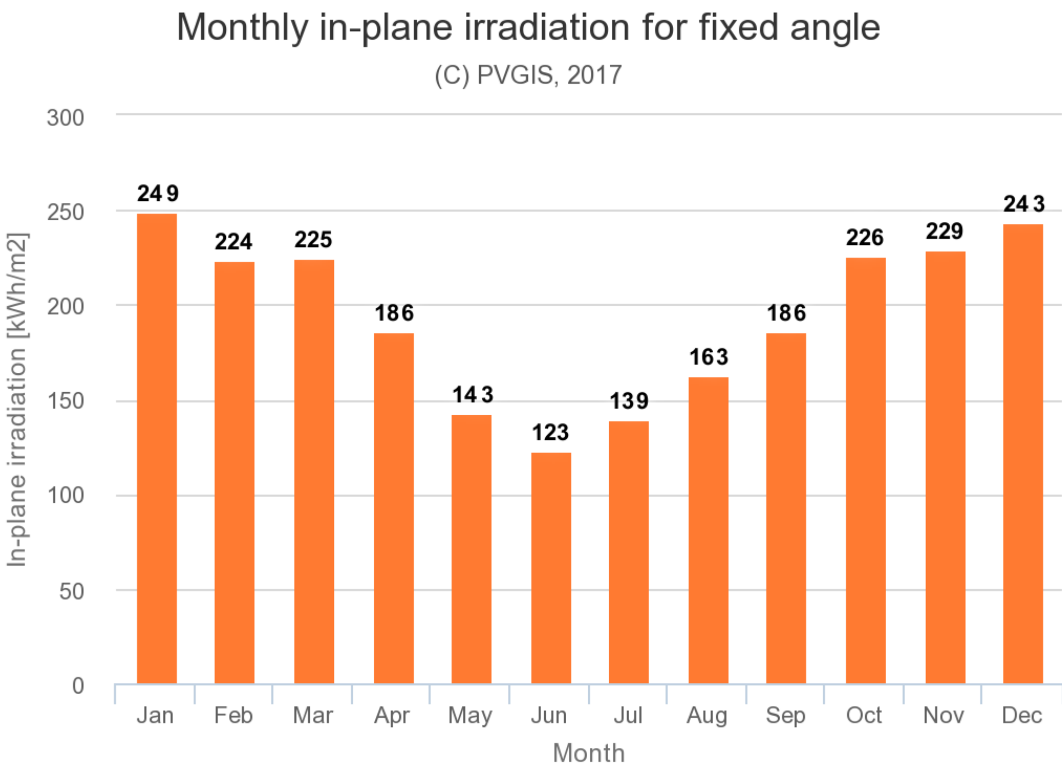

Figure 6.7: Monthly in-plane irradiation for fixed angle

In this thesis solarGis is considered as a global source of irradiation. Table 6.3 presents the global horizontal irradiation and temperature- climate reference in the selected coordinates; while Figure 6.8 represents these data- monthly irradiation, daily sum of irradiation, daily sum of diffuse irradiation, and daily air temperature.- in a graph.

Table 6.3: Global horizontal irradiation and air temperature

Month	Gh_m	Gh_d	Dh_d	T_{24}
Jan	256.6	8.28	1.65	21.1
Feb	207.8	7.42	1.47	21.5
Mar	183.5	5.92	1.40	20.3
Apr	128	4.27	1.21	18.2

May	89.3	2.88	1.10	16.1
Jun	72.6	2.42	0.93	14.2
Jul	83.8	2.70	0.98	13.5
Aug	105.8	3.41	1.28	13.6
Sep	143.6	4.79	1.66	14.8
Oct	197.5	6.37	1.82	16.7
Nov	226.7	7.56	1.92	18.2
Dec	254.6	8.21	1.92	20.0
Year	1949.8	5.34	1.44	17.3

Where:

Gh_m Monthly sum of global irradiation [kWh/m^2]

Gh_d Daily sum of global irradiation [kWh/m^2]

Dh_d Daily sum of diffuse irradiation [kWh/m^2]

T_{24} Daily air temperature [^0C]

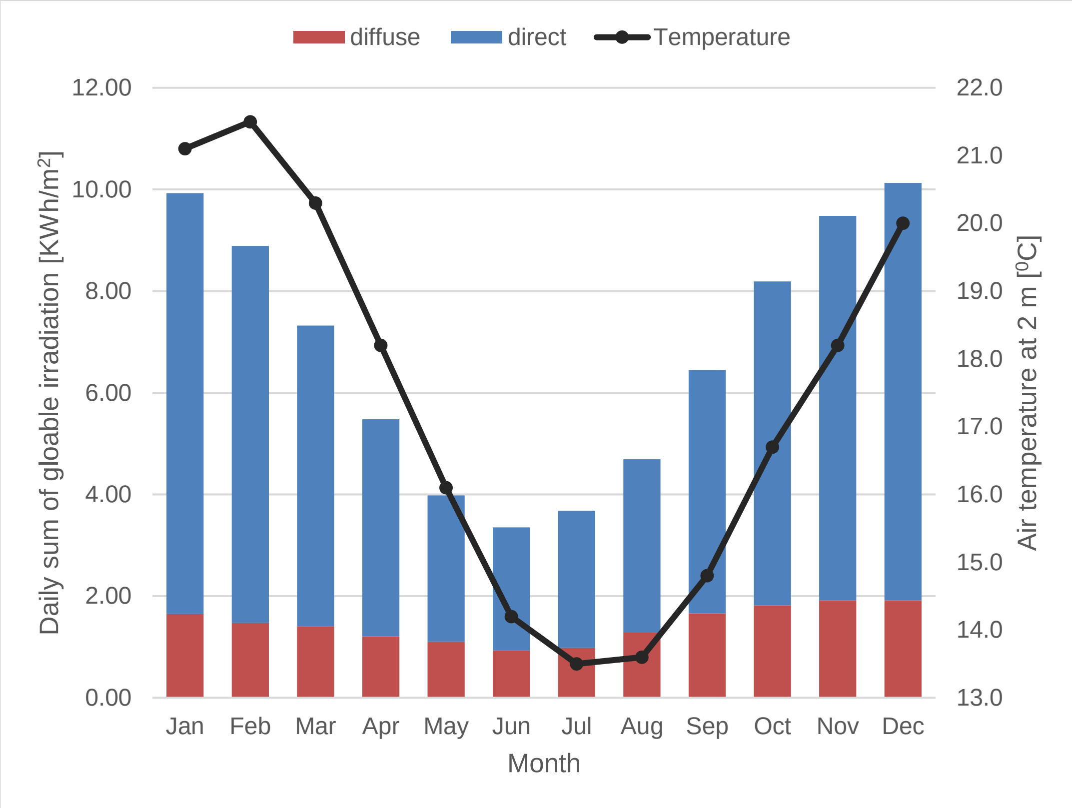

Figure 6.8: Average daily and monthly of global irradiation

6.8 Global in-plane irradiation

Fixed surface, azimuth 0° (north), inclination. 28°

Table 6.4: Global in-plane irradiation

Month	Gi_m	Gi_d	Di_d
Jan	239.8	7.74	1.65
Feb	213.1	7.62	1.55
Mar	214.1	6.91	1.56
Apr	170.1	5.67	1.40
May	129.7	4.18	1.27
Jun	112.5	3.75	1.10
Jul	127.1	4.10	1.16

Aug	143.2	4.62	1.45
Sep	173.1	5.77	1.81
Oct	210.9	6.80	1.90
Nov	218.1	7.27	1.91
Dec	231.8	7.47	1.86
Year	2183.5	5.98	1.55

Where:

Gi_m Monthly sum of global irradiation [kWh/m^2]

Gi_d Daily sum of global irradiation [kWh/m^2]

Di_d Daily sum of diffuse irradiation [kWh/m^2]

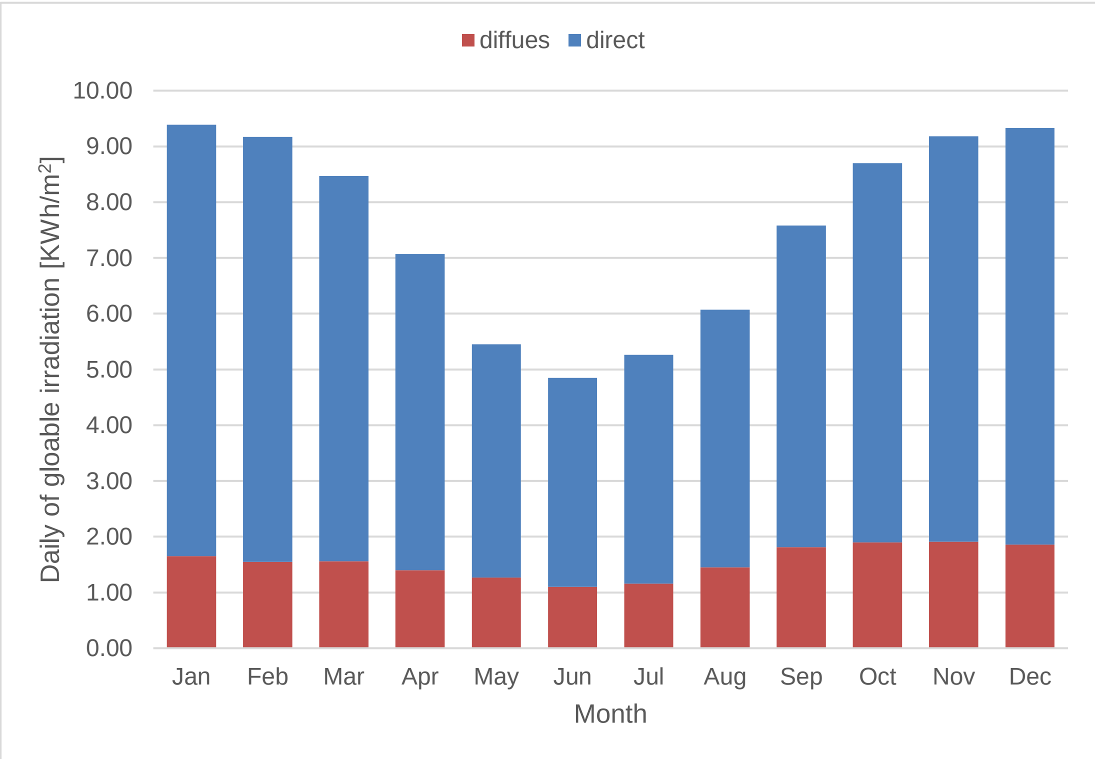

Figure 6.9: Global in-plane irradiation

6.9 PV electricity production in the start-up

Long-term monthly averages:

Es_m Monthly sum of specific electricity prod [kWh/kW$_p$]

Es_d Daily sum of specific electricity prod [kWh/kW$_p$]

Et_m Monthly sum of total electricity prod [kWh]

E_{share} Perceptual share of monthly electricity prod [%]

PR Performance ratio [%]

6.10 System losses and performance ratio

The energy conversion losses considered the following steps:

1. Initial production at Standard Test Conditions (STC) is assumed

2. Reduction of global in-plane irradiation due to obstruction of terrain horizon and PV modules

3. Proportion of global irradiation that is reflected by surface of PV modules (typically glass)

4. Losses in PV modules due to conversion of solar radiation to DC electricity; deviation of module efficiency from STC

5. DC losses: this step assumes integrated effect of mismatch between PV modules, heat losses in interconnections and cables, losses due to dirt, snow, icing and soiling, and self-shading of PV modules

6. This step considers to approximate average losses in the inverter,

7. Losses in AC section and transformer (where applicable) depend on the system architecture,

8. Availability parameter assumes losses due to downtime caused by maintenance or failures.

Table 6.5: System performance

Energy conversion step	Energy output	Energy loss	Energy loss	Performance ratio	
	[kWh/KW$_p$]	[kWh/kW$_p$]	[%]	[partial %]	[cumul. %]
Global in-plane irradiation (input)	2187	-	-	100.0	100.0
Global irradiation reduced by terrain shading	2184	-3	-0.1	99.9	99.9
Global irradiation reduced by reflectivity	2128	-56	-2.5	97.4	97.3
Conversion to DC in the modules	1922	-206	-9.7	90.3	87.9
Other DC losses	1816	-106	-5.5	94.5	83.0
Inverters (DC/AC conversion)	1771	-45	-2.5	97.5	81.0
Transformer and AC cabling losses	1744	-27	-1.5	98.5	79.8
Reduced availability	1727	-18	-1.0	99.0	79.0
Total system performance	1727	-460	-21.1	-	79.0

6.11 System overview

The design of a PV system was guided by various considerations such as the load profile, the site topography, as well as the inverter and module sizes. This section provides a brief overview of the preliminary design. Table 6.6 provides an overview of the preliminary design with the main features of the PV system and annual energy yield. The installed capacity is 89.96 kW$_p$, therefore it is expected that all of the produced energy will be consumed by the facility. The Solar PV system consists of 346 solar modules poly crystalline type connected to two 50 kW inverters. For optimum yields the system was been divided into two subprojects.

Table 6.6: Preliminary solar PV design

System Overview			
346 x SolarWorld AG SW 260 poly Protect (PV array 1) Azimuth angle: 180^0, Tilt angle: 28^0, Mounting type: Ground mount, peak power: 89.96 KW_p			
PV design data			
Peak power:	89.96 kWp	Annual energy yield*:	172.95 MWh
Total number of PV modules:	346	Energy usability factor:	100 %
Number of PV inverters:	2	Performance ratio*:	89.5 %
Nominal AC power of the PV inverters:	100.00 KW	Spec. energy yield*:	1922 kWh/kW_p
AC active power:	100.00 KW	Line losses (in % of PV energy):	
Active power ratio: 100.2 %	111.2 %		

Table 6.7: Subproject 1 Acacia PV water system

1×STP 50-40 (PV system section 1)	
Peak power	44.20 KW_p
Total number of PV modules:	170
Number of PV inverters:	1
Max. DC power (cos φ = 1):	51.00 kW

Max. AC active power (cos φ = 1):	50.00 KW
Grid voltage:	230V (230V/400V)
Nominal power ratio:	115 %
Dimensioning factor:	88.4 %
Displacement power factor cos φ:	1

Table 6.8: Subproject 1(PV design data)

Input A: PV array 1: 44 x SolarWorld AG SW 260 poly Protect, Azimuth angle: 180 °, Tilt angle: 28 °, Mounting type: Ground mount			
Input B: PV array 1: 44 x SolarWorld AG SW 260 poly Protect, Azimuth angle: 180 °, Tilt angle: 28 °, Mounting type: Ground mount			
Input C: PV array 1: 22 x SolarWorld AG SW 260 poly Protect, Azimuth angle: 180 °, Tilt angle: 28 °, Mounting type: Ground mount			
Input D: PV array 1: 44 x SolarWorld AG SW 260 poly Protect, Azimuth angle: 180 °, Tilt angle: 28 °, Mounting type: Ground mount			
Input E: PV array 1: 16 x SolarWorld AG SW 260 poly Protect, Azimuth angle: 180 °, Tilt angle: 28 °, Mounting type: Ground mount			
	Input A:	Input B:	Input C:
Number of strings:	2	2	1
PV modules per string:	22	22	22
Peak power (input):	11.44 KW_p	11.44 KW_p	5.72 KW_p
Typical PV voltage:	659 V	659 V	659 V
Min. PV voltage:	617 V	617 V	617 V
Min. DC voltage	150 V	150 V	150 V
Max. PV voltage	908 V	908 V	908 V
Max. DC voltage:	1000 V	1000 V	1000 V
Max. current of PV array:	16.7 A	16.7 A	8.4 A

Max. DC current:	20 A	20 A	20 A
Max. input short-circuit current per MPPT:	30 A	30 A	30 A
Photovoltaic Output Circuit Current:	17.9 A	17.9 A	8.9 A

	Input D:	Input E:
Number of strings:	2	1
PV modules per string:	22	16
Peak power (input):	11.44 KW_p	4.16 KW_p
Typical PV voltage:	659 V	480 V
Min. PV voltage:	617 V	449 V
Min. DC voltage	150 V	150 V
Max. PV voltage	908 V	661 V
Max. DC voltage:	1000 V	1000 V
Max. current of PV array:	16.7 A	8.4 A
Max. DC current:	20 A	20 A
Max. input short-circuit current per MPPT:	30 A	30 A
Photovoltaic Output Circuit Current:	17.9 A	8.9 A

Table 6.9: Subproject 2 Acacia PV water system

1×STP 50-40 (PV system section 2)	
Peak power	45.76 KW_p
Total number of PV modules:	176
Number of PV inverters:	1

Max. DC power (cos φ = 1):	51.00 kW
Max. AC active power (cos φ = 1):	50.00 KW
Grid voltage:	230V (230V/400V)
Nominal power ratio:	111 %
Dimensioning factor:	91.5 %
Displacement power factor cos φ:	1

Table 6.10: Subproject 2 (PV design data)

Input A: PV array 1: 44 x SolarWorld AG SW 260 poly Protect, Azimuth angle: 180 °, Tilt angle: 28 °, Mounting type: Ground mount				
Input B: PV array 1: 44 x SolarWorld AG SW 260 poly Protect, Azimuth angle: 180 °, Tilt angle: 28 °, Mounting type: Ground mount				
Input C: PV array 1: 44 x SolarWorld AG SW 260 poly Protect, Azimuth angle: 180 °, Tilt angle: 28 °, Mounting type: Ground mount				
Input D: PV array 1: 44 x SolarWorld AG SW 260 poly Protect, Azimuth angle: 180 °, Tilt angle: 28 °, Mounting type: Ground mount				
	Input A:	Input B:	Input C:	Input D:
Number of strings:	2	2	2	2
PV modules per string:	22	22	22	22
Peak power (input):	11.44 KW_p	11.44 KW_p	11.4 KW_p	11.4 KW_p
Typical PV voltage:	659 V	659 V	659 V	659 V
Min. PV voltage:	617 V	617 V	617 V	617 V
Min. DC voltage	150 V	150 V	150 V	150 V
Max. PV voltage	908 V	908 V	908 V	908 V
Max. DC voltage:	1000 V	1000 V	1000 V	1000 V
Max. current of PV array:	16.7 A	16.7 A	16.7 A	16.7 A
Max. DC current:	20 A	20 A	20 A	20 A
Max. input short-circuit current per MPPT:	30 A	30 A	30 A	30 A

Photovoltaic Output Circuit Current:	17.9 A	17.9 A	17.9 A	17.9 A

This design was based on various assumptions and considerations and only serves as a preliminary solution and is subject to detailed designs and confirmation. The return on investment was subject to the cash flow of the project, which is dependent of numerous variables. The variance and uncertainty associated with each of these variables contributes to the perceived risk of the investment. However, by accounting for these parameters, the risk of the investment can be quantified and minimised.

The income of a PV project is the product of the annual yield of the PV plant and the electricity tariff. The annual output of the system decreases over time, while the tariff is expected to increase yearly. The highest expense associated with the project is the capital cost of the PV system. Although the operation and maintenance costs are almost negligibly small, there may be some replacement costs during the project life.

6.12 Summary

Recently, the importance of solar PV power systems has attracted the attention of researchers because solar energy is a free, efficient, clean, and an environmentally renewable and sustainable energy resource.

In this chapter an overview of Solar PV system components, system design principles, and the values and limits of system parameters for assessing system performance, were provided. This data is intended for those individuals who specify solar PV systems equipment and evaluate system designs, as well as those who design and integrate systems. This chapter contains a detailed explanation of the methodology that was implemented to make this project feasible.

According to the information contained in this project it is evident that the proposed solar PV system is technically feasible and is expected to be a suitable strategy to offset the requirement of the annual electricity consumption of the cooling facility from the utility grid. A $90kW_p$ PV system would minimise the parasitic demand and maximise the return on investment. The capital expenditure of approximately R1,800,000.00 is expected to be recovered within 7 to 10 years. Therefore, the proposed PV investment can be considered as a hedging mechanism against the rising cost of energy and to improve the competitive advantage of the enterprise.

CHAPTER SEVEN

CONCLUSION AND RECOMMENDATIONS

7 CONCLUSION AND RECOMMENDATIONS

7.1 Introduction

The main goal of this dissertation was to develop, design and study the performance of Inlet fogging cooling system to increase the overall power generation of gas turbine of Acacia power plant in hot environmental temperatures. To further increase the efficiency, a Solar PV system was adopted and designed to supply the power requirements for the inlet fogging cooling facility.

In the first chapter a general introduction to the subject of the project was presented. This chapter further explains the scope of this thesis and formulates the problems that were discussed in the manuscript.

In chapter two of this thesis, a comprehensive literature review of gas turbines inlet cooling technologies was presented with particular attention to the state-of-the-art of inlet fogging systems. This chapter intended to give a general background of different available inlet cooling technologies in the industry. Comparison and case studies of different application of this technology was also presented.

Chapter three, presents an overview of a simple gas turbine and its components was presented. Details on the thermodynamics cycle was provided. A derivation of thermal efficiency with regards to inlet temperature was delivered. Included in this chapter are the discussions on the performance of the gas turbine with inlet temperature. The chapter also contains gas turbines energy production share in South Africa and a brief introductory as to why the Acacia power plant was selected.

In chapter four with regards to study of the inlet cooling operating applications, an overview of gas turbines inlet air cooling with definition of each technology was presented. The key benefits and drawbacks of various technologies were highlighted. Also, the economic and environmental perspective of inlet air cooling was given. Lastly, the analyses of GT performance with ambient temperature and humidity were demonstrated.

Chapter fives of this thesis is dedicated to the-state-of-the-art of fogging system design layout. This was the core of this project, where technical aspects of the fogging system design were expounded. Details on water quality and nozzles arrangement consideration and requirements were described in this chapter. The design condition and the arrangement of the fogging system were provided. The excepted impact of the cooling on the selected gas turbine was also examined.

In chapter six, solar PV design overview and load profile analysis was presented. Also assessments were presented of the photovoltaic PV system design and yield of the PV power. The average daily and monthly of global irradiation were obtained.

In this final chapter conclusion and further recommendations are drawn.

7.2 Conclusion

This work intends to provide the technical information requirements for the inlet fogging cooling system design to achieve power output augmentation for Acacia power plant located in Cape Town, South Africa. Some technical data were not received from the plant's operator. However, calculations were based on some necessary assumptions that have been assessed with the cooperation of the American Moistening Company (AMCO). Nevertheless, the results of this work showed the ability of inlet fogging system to cool the inlet air during periods of hot weather. Hence, providing the gas turbine to operate more efficiently with increased of power output approximately up to 5.9%. The fogging system expected to reduce the intake compressor air temperature by 5.7^{0}C for a 2.2^{0}C approach to the design point wet bulb temperature. This can decrease the gas emission and heat rate per unit m MWs. Additional power output gains of 11.4% and 16.9% of the rated power can be achieved if 1% and 2% overspray systems implemented respectively as proposed in the design.

In this study, a 204 bar was selected in consultation with a well-known inlet fogging manufacturer AMCO. The units of the selected power plant were studied in order to increase the power augmentation output in hot weather conditions. The performance of the inlet fogging system was analysed and illustrated.

The use of solar PV energy is becoming increasingly attractive due to high reliability, low maintenance, no moving parts, low running costs, and the long life expectancy of the main components. The proposed work herein was on the development of a photovoltaic (PV) application for driving the fogging system. From the estimated values as well as real data simulation, it is evident that the proposed solar PV system is technically valuable and is expected to compensate the requirement of the annual electricity of the inlet fogging cooling facility's electricity consumption. A 90 kW_p PV system would contribute partially not just for power self-consumption but also to maximize the return on investment. Although the high capital cost of the project, including engineering procurement and construction is high, the capital expenditure is expected to be recovered by the energy savings generated by the system within 7 to 10 years. To conclude, the following remarks could be made:

1. The results indicated that the inlet air cooling technologies are primordial to ensure the gas turbine stability production in hot conditions.

2. The study indicated that the solar PV system was able to supply adequately enough power demands of the inlet cooling fogging system.

3. It is vital to understand that the additional components required of any cooling system will add additional cost of the power plant. However, these added components present an inferior cost when compared with a large simple-cycle gas turbine engine.

4. The analyses showed that efficiency and power output decrease with an increase in ambient temperature. Hence, ambient temperature plays a major role on Acacia gas turbines power generation.

5. The importance of water quality cannot be overstated, particularly when fogging system is required.

6. From an economical and technical perspective, the benefit of fogging appears to be threefold: lower capital cost, more effective cooling, and a much lower gas turbine inlet pressure drop.

7. Various factors contributed in making the decision of best cooling technology method such as annual local temperature, relative humidity, water availability, construction cost, operation cost and periodic maintenance.

8. In isolated hot areas with high levels of radiation making use of solar PV energy to supply inlet cooling system power requirements is a sustainable approach.

7.3 Recommendations

Further recommendations can be mentioned of this project to extend the feasibility assessment and to enhance the performance in several directions. The following are some recommendations for future work:

1. Review other types of gas turbine inlet cooling technologies and present in detail their advantages and disadvantages.

2. For better usage of fogging system, we recommend the technique is best used for cooling gas turbines in hot with relatively less humid conditions.

3. The water quality requirements vary for each of GTIC methods and should be considered as it can result in added operational costs to the turbine user.

4. The effect of humidity and water droplets on compressor blades erosion and hence performance is required further studies.

5. Operation and Maintenance (O&M) cost must also be considered, and varies considerably with the type of CTIC.

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APPENDICES

Appendix A: Partial Database of Turbine Inlet Cooling (TIC) Installation

Updated: September 8, 2016

Notes: All data are approximate and represent examples of TIC installations; however, values reported for each TIC technology are not necessarily representative of the actual number of installations nor are they indicative of the total number for each cooling technology. Efforts have been made to verify the accuracy of the data; however, TICA makes no warranty regarding accuracy or completeness. Data were obtained primarily from TICA members and from other published sources.

	Turbine Installation Data								Hot Weather Power Enhancement from TIC [1]		TIC System Developer, Designer, Equipment Supplier(s), or Installer	
Initial Year of TIC Operation	CT Plant Owner / Operator	CT Plant Location	Simple or Combined Cycle (SC or CC)	TIC Applied to Existing or New CTs	Quantity of CTs	CT Make	CT Model	CT Plant I.S.O. Output (MW)	TIC Power Increase (MW)	TIC Power Increase (%)	**TICA Member with Primary Involvement**	**Other TICA Member(s) Contributing Products or Services to the Project**
2018	Dominion Greensville County	Virginia, USA	CC	New	3	MHI	501J	1,354.0	168.0	12.0%	Turbine Air Systems	CB&I
2017	Gulf SPP GTS2	Thailand	CC	New	2	Siemens	SGT-800B	120.5	17.6		Turbine Air Systems	
2017	Gulf SPP GTS1	Thailand	CC	New	2	Siemens	SGT-800B	120.5	17.6		Turbine Air Systems	
2017	Gulf SPP GVTP	Thailand	CC	New	2	Siemens	SGT-800B	120.5	17.6		Turbine Air Systems	
2017	HF Lee CC	North Carolina, USA	CC	New	2	Siemens	SGT6-5000F	746.0	81.0		Turbine Air Systems	
2017	GREC 3	Oklahoma, USA	CC	New	1	MHI	501J	452.0	60.0		Turbine Air Systems	
2016	Malacas	Peru	SC	New	1	Siemens	SGT-800	46.0	4.5		Turbine Air Systems	
2016	Dominion Brunswick County	Virginia, USA	CC	New	3	MHI	501GAC	1,329.0	107.5	8.8%	Turbine Air Systems	CB&I
2015	Batangas	Philippines	SC	New	2	LM 6000 PC-S					Turbine Air Systems	
2015	Baytown	Texas, USA	CC	Existing	3	W501FD					Turbine Air Systems	
2015	G LNG Train 2	Australia	SC	New	6	LM 2500+G4					Turbine Air Systems	
2015	G LNG Train 1	Australia	SC	New	6	LM 2500+G4					Turbine Air Systems	
2015	AP LNG Train 2	Australia	SC	New	6	LM 2500+G4					Turbine Air Systems	
2015	AP LNG Train 1	Australia	SC	New	6	LM 2500+G4					Turbine Air Systems	
2015	QC LNG	Australia	SC	New	12	LM 2500+G4					Turbine Air Systems	
2014	Amata B. Grim 4 & 5	Thailand	CC	New	4	Siemens	SGT800	246.0	24.0		Turbine Air Systems	
2014	Ibese	Nigeria	SC	New	2	G.E.	LM 6000 PC-S	96.0	13.0		Turbine Air Systems	
2014	Obajana	Nigeria	SC	New	1	G.E.	LM 6000 PC-S	48.0	10.0		Turbine Air Systems	
2013	Bulo Bulo	Bolivia	SC	New	1	G.E.	LM 6000 PC	42.0	9.0		Turbine Air Systems	
2013	Sunshine Canyon	California, USA	SC	New	5	Solar	Mercury 50	23.0	4.0		Turbine Air Systems	
2013	Tihama	Saudi Arabia	SC	New	3	G.E.	7FA	544.0	41.0		Turbine Air Systems	
2013	Nesher Cement	Israel	CC	New	2	G.E.	LM 6000 PF	135.0	24.0		Turbine Air Systems	
2013	Golden Spread	Texas, USA	SC	Existing	1	G.E.	7FA	172.0	12.0		Turbine Air Systems	
2013	Dominion Warren County	Virginia, USA	CC	New	3	MHI	501GAC	1,329.0	107.5	8.8%	Turbine Air Systems	DN Tanks
2012	Diamantina	Australia	CC	New	4	Siemens	SGT800	242.0	24.0		Turbine Air Systems	
2012	Solomon I	Australia	SC	New	2	G.E.	LM6000PF	85.0	24.0		Turbine Air Systems	
2012	SWES Ghana	Ghana	CC	New	4	Orenda	GT25000	100.0	19.0		Turbine Air Systems	
2012	Proctor and Gamble	Mehoopany, PA, USA	CC	New	1	Rolls Royce	Trent 60	51.0	14.0	27.5%	Stellar	
2012	Diamond Generating Corp.	Mariposa, CA, USA	CC	New	4	G.E.	LM 6000 PC-S	184.0	54.0	29.3%	Stellar	
2011	University of Texas	Austin, TX, USA	CC / CHP	New/Exist	2			32 + 45	4 + 5	14.0%	CB&I	Cool Solutions
2011	Talang Duku	Indonesia	SC	New	2	G.E.	TM 2500	62.0	5.0		Turbine Air Systems	
2011	Morichal	Venezuela	SC	New	2	G.E.	LM6000 PC-S	87.4	12.6	14.5%	Turbine Air Systems	
2011	La Raisa II	Venezuela	SC	New	2	G.E.	LM6000 PC-S	86.2	13.8	16.1%	Turbine Air Systems	
2011	Dan River	North Carolina, USA	CC	New	2	G.E.	7FA	620.0	53.0		Turbine Air Systems	
2011	Amata B. Grim	Thailand	CC	New	2	Siemens	SGT800A	123.0	12.0		Turbine Air Systems	
2011	SNC Lavalin	Peru	CC	New	2	G.E.	7241 FA	370.0	86.0	23.2%	Stellar	
2011	Petrobras	Brazil	CC	New	1	G.E.	LM 6000 PC-S	46.0	12.0	26.1%	Stellar	
2011	SG Petroleum	Kuwait City, Kuwait	SC	Existing	2	G.E.	LM 6000 PC	84.2	43.3	105.6%	Turbine Air Systems	
2010	TECO	Houston, TX, USA	SC / CHP	New	1			45.0			CB&I	Cool Solutions
2010	Songas	Tanzania	SC	Existing	3	G.E.	LM 6000 PC	102.9	15.5	12.7%	Turbine Air Systems	
2010	Black Hills Colorado IPP	Colorado, USA	CC	New	4	G.E.	LM 6000 PC-S	184.0	42.0	22.8%	Stellar	
2010	Black Hills / Colorado Electric	Colorado, USA	CC	New	2	G.E.	LMS 100 PA	196.0	36.0	18.4%	Stellar	
2010	Dominion Energy - Bear Garden	New Canton, VA, USA	CC	New	2	G.E.	PG 7241 FA	560.0	60.3	13.5%	Turbine Air Systems	DN Tanks
2010	City of Anaheim	Anaheim, CA, USA	SC	New	4	G.E.	LM 6000 PC-S	185.1	34.8	20.9%	Turbine Air Systems	

	Turbine Installation Data								Hot Weather Power Enhancement from TIC [1]		TIC System Developer, Designer, Equipment Supplier(s), or Installer	
Initial Year of TIC Operation	CT Plant Owner / Operator	CT Plant Location	Simple or Combined Cycle (SC or CC)	TIC Applied to Existing or New CTs	Quantity of CTs	CT Make	CT Model	CT Plant I.S.O. Output (MW)	TIC Power Increase (MW)	TIC Power Increase (%)	TICA Member with Primary Involvement	Other TICA Member(s) Contributing Products or Services to the Project
2010	GenConn Middletown, LLC	Middletown, CT, USA	SC	New	4	G.E.	LM 6000 PC-S	185.1	29.9	17.9%	Turbine Air Systems	
2010	GenConn Devon, LLC	Milford, CT, USA	SC	New	4	G.E.	LM 6000 PC-S	185.1	30.2	18.1%	Turbine Air Systems	
2010	Coolidge Power	Arizona, USA	SC	New	12	G.E.	LM 6000	576.0	htg only		Turbine Air Systems	
2010	Enmax Green Power	Calgary, AB, Canada	SC	New	3	G.E.	LM 6000	144.0	htg only		Turbine Air Systems	
2010	Buck Station	North Carolina, USA	CC	New	2	G.E.	7FA	550.0	48.0		Turbine Air Systems	
2010	Austin Energy	Austin, TX, USA	SC	New	2	G.E.	LM 6000 PC-S	92.6	24.5	33.4%	Turbine Air Systems	
2010	Brazos Electricl Coop - Johnson I	Cleburne, TX, USA	CC	Existing	1	Siemens	501 F	250.0	35.9	15.3%	Turbine Air Systems	DN Tanks
2009	Colorado Energy Management	Hobbs, NM, USA	CC	New	2	MHI	501 FD2	188.0	19.0	10.1%	Stellar	
2009	Brazos Electric Coop - Jack I & II	Jacksboro, TX, USA	CC	Exist+New	2 + 2	G.E.	PG 7241 FA	1,120.0	101.2	11.0%	Turbine Air Systems	DN Tanks
2009	Confidential	California, USA	SC	New	2	G.E.	LM 6000 PC-S	92.6	0.0	#DIV/0!	Turbine Air Systems	
2009	Mackinaw Power LLC	Southeast USA	CC	New	2	G.E.	PG 7241 FA	560.0	48.4	10.9%	Turbine Air Systems	
2009	Topaz - Barney Davis	Texas, USA	CC	New	2	G.E.	PG 7241 FA	500.0	51.4	11.5%	Turbine Air Systems	
2009	Topaz - Nueces Bay	Texas, USA	CC	New	2	G.E.	PG 7241 FA	500.0	51.4	11.5%	Turbine Air Systems	
2009	City Public Service	Elmendorf, TX, USA	SC	Existing	2	G.E.	7FA				Stellar	
2009	City Public Service	Elmendorf, TX, USA	SC	New	4	G.E.	LM6000				Stellar	
2009	Western Farmers Electric Cooperative	Anadarko, OK, USA	SC	New	3	G.E.	LM6000				Stellar	
2009	Southern Co.	USA	CC	Existing	2	G.E.	7FA				Munters	
2009	FP&L	USA	CC	Existing	6	G.E.	7FA				Munters	
2009	FP&L	USA	CC	Existing	3	G.E.	7FA	750.0			Munters	
2009	Dominion Energy - Fairless Hills Ph 2	Fairless Hills, PA, USA	CC	New	4	G.E.	PG 7241 FA	1,038.0	114.9	12.9%	Turbine Air Systems	CB&I, Cool Solutions
2009	BP Rodeo	Texas, USA	CC	New	1	Solar	Mercury 50	4.0	1.0		Turbine Air Systems	
2009	Tampa Electric	USA			5	PWPS	FT8				Munters	
2008	Arizona Public Service	Arizona, USA	SC	New	2	G.E.	LM 6000 PC-S	92.6	15.6	19.2%	Turbine Air Systems	
2008	ABA	Africa	SC	New	3	G.E.	LM 6000 PC-S	139.2	13.8	10.5%	Turbine Air Systems	
2008	Shumaik	Kuwait	SC	New	3	G.E.	LM 6000 PC-S	139.2	35.1	35.7%	Turbine Air Systems	
2008	Alghanim	Kuwait	SC	New	6	G.E.	LM 6000 PC-S	278.4	74.7	38.0%	Turbine Air Systems	
2008	Dominion Energy - Fairless Hills Ph 1	Fairless Hills, PA, USA	CC	New	2	G.E.	PG 7241 FA	519.0	57.5	12.9%	Turbine Air Systems	
2008	L'Energia Power Station	Massachusetts, USA	SC	New	1	Rolls Royce	Trent 60	50.0	10.5	22.9%	Turbine Air Systems	
2008	Niland / Imperial Irrigation District	California, USA	SC	New	2	G.E.	LM 6000 PD-S	92.8	27.1	37.2%	Turbine Air Systems	GEA
2008	Pacific Gas & Electric Company	California, USA	CC	New	2	G.E.	PG 7241 FA	528.0	61.0	14.1%	Turbine Air Systems	GEA
2008	Winchester Peakers	Texas, USA	SC	New	4	G.E.	LM 6000 PD-S	185.6	34.2	21.7%	Turbine Air Systems	GEA
2008	Uruguay UTE Plant	South America	SC	New	2	G.E.	LM 6000 PC-S	92.8	2.3	2.4%	Turbine Air Systems	
2008	Akmaya	Turkey		Existing	2	Kawasaki	GTC70A	14.0			Munters	
2008	Zorlu Energy	Turkey		Existing		ACC units					Munters	
2008	Mopak	Turkey		Existing	1	Solar	Taurus 60	5.0			Munters	
2008	Entek	Turkey		Existing	5	G.E.	LM6 & 2500	174.0			Munters	
2008	Aksa	Turkey		Existing	4	G.E.	LM6000	188.0			Munters	
2008	Ak Gida	Turkey		Existing	1	Solar	Taurus 70	7.0			Munters	
2008	Besler Gida	Turkey		Existing	3	Solar	Taurus 60	15.0			Munters	
2008	Bosen Energy	Turkey		Existing	2	G.E.	LM6000	84.0			Munters	
2008	Enterprise	USA		Existing	2	G.E.	Frame 5	50.0			Munters	
2008	Packerab/Ge Oil & Gas	Middle East		New	4		compres'r dr	40.0			Munters	
2008	Marib/Ge Oil & Gas	Middle East		New	1		compres'r dr	25.0			Munters	
2008	Neerabup	Australia		New	2		SGT5 2000E	370.0			Munters	
2008	Braemer	Australia		New	3		SGT5 2000E	555.0			Munters	
2008	Antalya	Turkey		New	2		SGT5 4000F	550.0			Munters	
2008	Drewsen	Germany		Existing	5	Solar	Taurus 70	5.0			Munters	
2008	Quatalum	Quatar		New	4	G.E.	Frame 9FA	1,020.0			Munters	
2008	Garri Power Plant	Sudan		Existing	8	G.E.	Frame 6	320.0			Munters	
2008	HECO	USA		New	1	Siemens	501D	136.0			Munters	
2008	Arsenal Hill	USA		New	2	Siemens	501F	373.0			Munters	

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2007	DCP Midstream/Pegasus	US		Existing	2	Solar		10.0			Munters	
2007	Evander Andrews	US		New	1	Siemens	501F	372.0			Munters	
2007	Caithness	US		New	1	Siemens	501F	372.0			Munters	
2007	Amylum Nisasta	Turkey		Existing	1	Solar	Titan	15.0			Munters	
2007	Hayat Temizlik	Turkey		Existing	2	Solar	Taurus 70	11.0			Munters	
2007	Pakmaya	Turkey		Existing	3	Solar	Taurus 60	15.0			Munters	
2007	Alstom (G&H) Munmorah	Australia		New	4	ABB	GT 13 E2	720.0			Munters	
2007	Alstom (R&M) Rio TKS	Brazil		New	2	ABB	GT 11 N2	228.0			Munters	
2007	Alstom (R&M) Al Zhour	Kuwait		New	5	ABB	GT 13 E2	900.0			Munters	
2007	Siemens (R&M) Jebel Ali M	UAE		New	6	Siemens	V 94.3A	1,650.0			Munters	
2007	Williams Energy	Turkey		Existing	2	Solar	Taurus 70	11.0			Munters	
2007	Williams Energy	Turkey		Existing	3	Solar	Taurus 60	15.0			Munters	
2007	Uranquinty	Australia	SC	New	4	Siemens	V94.2	640.0			Munters	
2007	Williams Energy	US		Existing	2	NG compres'r		10.0			Munters	
2007	Mesaieed GE	Qatar		New	6	G.E.	Frame 9FA	1,530.0			Munters	
2007	Southern California Energy	US		New	2	G.E.	LM 6000	86.0			Munters	
2007	Termozulia	Venezuela		New	2	Westinghouse	501F	372.0			Munters	
2007	Kimbassan	Turkey		Existing	1	Solar	Taurus	5.0			Munters	
2007	Halkali Kagit	Turkey		Existing	1	Solar	Taurus	5.0			Munters	
2007	Cyco Fos	France	CC	New	1	ABB	GT 26B	420.0			Munters	
2007	Tallawara	Australia	CC	New	1	ABB	GT 26B	400.0			Munters	
2007	confidential owner	US		Existing	2	Westinghouse	501F	372.0			Munters	
2007	Reliance Industries Limited	Patalganga, Mah., India	SC / CHP	Existing	2	G.E.	MS 6001B	76.3	0.0	#DIV/0!	Cool Solutions	Avalon Consulting, Pasteris
2007	Sharikat Kahraba Hadjret En-Nouss	Wilaya of Tipaza, Algeria	CC	New	3	G.E.	9FB	1,227.0	160.0	#DIV/0!	Stellar	
2007	Inland Empire	California, USA	CC	New	2	G.E.	7H				Turbine Air Systems	
2006	Citizens Utilities Co	US		Existing	1	G.E.	LM2500	29.0			Munters	
2006	Pneumafil/Desert Basin	US		Existing	1	Siemens	501F	588.0			Munters	
2006	Pneumafil/Lakeside	US		Existing	2	Siemens	501F	373.0			Munters	
2006	Ege Seramik	Turkey		Existing	2	Solar	Centaur 50	8.0			Munters	
2006	Graniser	Turkey		Existing	1	Solar	Taurus 60	5.0			Munters	
2006	Termal Seramik	Turkey		Existing	1	Solar	Centaur 50	4.0			Munters	
2006	Altinyildiz	Turkey	CC	Existing	1	Solar	Taurus 60	5.0			Munters	
2006	Energetica Kladno	CZ		Existing	1	ABB	GT 8C	54.0			Munters	
2006	Stora Enso	Germany		Existing	1	G.E.	Frame 5	25.0			Munters	
2006	ENEL	Italy		New	2	Siemens	V94.3A	540.0			Munters	
2006	Siemens (R&M)	US		New	1	Siemens	5000 F	200.0			Munters	
2006	Siemens (R&M)	Middle East		New	3	Siemens	V94.3A	810.0			Munters	
2006	Siemens (R&M)	Middle East		New	4	Siemens	V94.3A	1,080.0			Munters	
2006	Kastamanou Entegre	Turkey		Existing	2	Solar	Taurus 60	10.0			Munters	
2006	Lenzing	Austria		Existing	2	Solar	Taurus 60	9.0			Munters	
2006	Goodyear	Turkey		Existing	2	Solar	Taurus 60	10.0			Munters	
2006	Alstom (G+H)	Australia		New	2	ABB	GT 26B	540.0			Munters	
2006	Siemens (G+H)	India		New	3	Siemens	V94.3A	840.0			Munters	
2006	Kappa Zülpich	Germany		Existing	3	Rolls Royce		14.0			Munters	
2006	Kartonsan	Turkey		Existing	4	Solar	Taurus 60	20.0			Munters	
2006	Hayat Kimya	Turkey		Existing	1	Solar	Taurus 60	5.0			Munters	
2006	Kastamanou Entegre	Turkey		Existing	1	Solar	Taurus 70	8.0			Munters	
2006	First Gas & Power	Phillipines		Existing	6	Siemens	V94.2A	936.0			Munters	
2006	Kwinana	Australia		New	1	ABB	GT 26	270.0			Munters	

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2005	Altair/GSEG	India		Existing	2	ABB	GT 8C	108.0			Munters	
2005	AAF/Unisource	USA		Existing	1	G.E.	LM2500	29.0			Munters	
2005	AAF/Iberese	USA		Existing	1	G.E.	LM2500	29.0			Munters	
2005	Pneumafil/Mankato	USA		Existing	1	Siemens	501F	187.0			Munters	
2005	Siemens (R&M)	Italy		New	2	Siemens	V94.3A	540.0			Munters	
2005	Alkim Kagit	Turkey		Existing	1	Solar	Taurus 60	5.0			Munters	
2005	Siemens (Kaefer)	Singapore		Existing	2	Siemens	V94.3A	540.0			Munters	
2005	ENEL	Italy		Existing	1	Siemens	V94.3A	270.0			Munters	
2005	Hayat Kagit	Turkey		Existing	1	Solar	Taurus 70	8.0			Munters	
2005	Desa	Turkey		Existing	2	Solar	Taurus 60	10.0			Munters	
2005	Ayka Tekstil	Turkey		Existing	1	Solar	Taurus 60	5.0			Munters	
2005	Tuma Turbomach	Pakistan		New	1	Solar	Taurus	5.0			Munters	
2005	Alstom (AAF)	Australia		New	3	ABB	13 E2	495.0			Munters	
2005	Tuma Turbomach	Switzerland		New	1	Solar	Taurus 60	5.0			Munters	
2005	Alstom (R&M)	Thailand		New	4	ABB	GT 26B	1,080.0			Munters	
2005	Man Turbo	Germany		Existing	2	FT8	Twinpack	96.0			Munters	
2005	Siemens (R&M)	Middle East		New	10	Siemens	V94.3A	1,600.0			Munters	
2005	Siemens (R&M)	Middle East		New	3	Siemens	V94.3A	780.0			Munters	
2005	ENEL	Italy		New	1	Siemens	V94.3A	260.0			Munters	
2005	Kings River Conservation District	Fresno, CA, USA	SC	New	2	G.E.	LM 6000	97.0	18.0	#DIV/0!	Stellar	
2005	Silicon Valley Power	San Jose, CA, USA	CC	New	2	G.E.	LM 6000	97.0	18.0	#DIV/0!	Stellar	
2005	Al Mussiab, Iraq	S. of Baghdad, Iraq	SC	New	10	G.E.	LM 6000	450.0				
2005	Austin Energy - Children's Hospital	Austin, TX, USA	SC / CHP	New	1	Solar	Mercury 50	4.3			Turbine Air Systems	CB&I, Cool Solutions
2005	confidential owner	Colombia	SC	New	2	G.E.	LM 6000	96.9	20.1	#DIV/0!	Turbine Air Systems	
2005	confidential owner	municipality, S.E. USA	SC	New	1	G.E.	LM 6000	42.6	14.5	#DIV/0!	Turbine Air Systems	GEA
2005	confidential owner	municipality, S.E. USA	SC	New	1	G.E.	LM 6000	42.6	14.5	#DIV/0!	Turbine Air Systems	GEA
2005	confidential owner	Nigeria	SC	New	3	G.E.	LM 6000	145.3	23.9	#DIV/0!	Turbine Air Systems	
2005	confidential owner	Riyadh, K. Saudi Arabia	SC	Existing	10	G.E.	7EA	750.0		30%	Stellar	Cool Solutions
2005	City of Lafayette	Lafayette , LA, USA	SC	New	2	G.E.	LM 6000	96.9	14.7	#DIV/0!	Turbine Air Systems	
2005	Modesto Irrigation District	Ripon, CA, USA	SC	New	2	G.E.	LM 6000	96.9	16.1	#DIV/0!	Turbine Air Systems	
2005	Princeton University	Princeton, NJ, USA	SC / CHP	Existing	1	G.E.	LM 1600	14.6			Cool Solutions	CB&I
2005	City of Riverside	Riverside, CA, USA	SC	New	2	G.E.	LM 6000	96.9	24.4	#DIV/0!	Turbine Air Systems	

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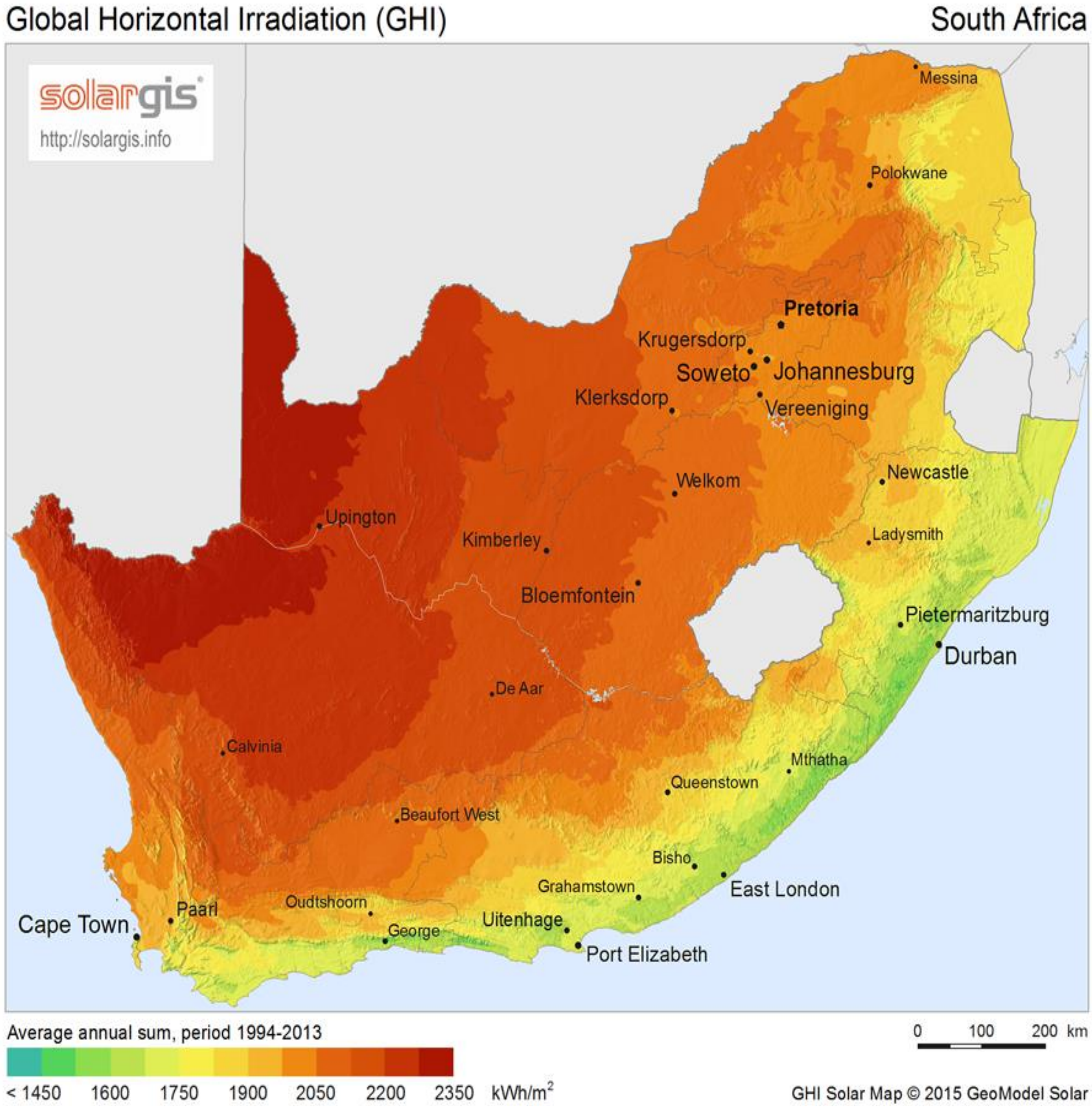
2004	Csepel Energia Kft	Hungary		Existing	2	G.E.	Frame 9E	266.0			Munters	
2004	Alstom (G+H)	India		New	2	ABB	13 E2	324.0			Munters	
2004	Akenerji	Turkey		Existing	6	EGT	Typhoon	30.0			Munters	
2004	Camfil	Greece		New	1	ABB	GT 10	19.0			Munters	
2004	Zorlu Enerji	Turkey		Existing	3	G.E.	LM 6000	126.0			Munters	
2004	Dresden Papier	Germany		Existing	2	Solar	Taurus 60	9.0			Munters	
2004	ENEL	Italy		Existing	9	Siemens	V94.3A	2,340.0			Munters	
2004	Modern Enerji	Turkey		Existing	2	ABB	GT 10	40.0			Munters	
2004	Modern Enerji	Turkey		Existing	2	gen'r cooling					Munters	
2004	Modern Enerji	Turkey		Existing	2	Solar	Mars	20.0			Munters	
2004	EEE	Turkey		Existing	1	G.E.	Frame 6	40.0			Munters	
2004	Bilenerji	Turkey		Existing	1	Rolls Royce	RB211	24.0			Munters	
2004	Austin Energy - Domain	Austin, TX, USA	SC / CHP	New	1	Solar	Centaur 50	4.5			Turbine Air Systems	
2004	confidential owner	Colombia	SC	New	1	G.E.	LM 6000	48.4	10.0	#DIV/0!	Turbine Air Systems	
2004	GFS	Long Island, NY, USA	SC	New	1	G.E.	LM 6000	48.4	7.6	#DIV/0!	Turbine Air Systems	
2004	Irag MOE	Iraq	SC	New	4	G.E.	LM 6000	170.5	58.9	#DIV/0!	Turbine Air Systems	
2004	Irag MOE	Iraq	SC	New	1	G.E.	LM 6000	42.6	14.1	#DIV/0!	Turbine Air Systems	
2004	Lafarge Gypsum Division	Silver Grove, KY, USA	SC / CHP	New	1			5.0				
2004	National Institute of Health	Bethesda, MD, USA	CC	New	1	Alstom	GT 10	22.0	2.7	14%		
2004	Newcrest Mining - Telfer	Port Hedland, Australia	SC	New	2	G.E.	LM 6000	96.8	0.0	#DIV/0!	Turbine Air Systems	
2004	NRG - Meriden [5]	Meriden, CT, USA	CC	New	2	G.E.	PG7241FA	475.0	64.4	#DIV/0!	Turbine Air Systems	
2004	NRG - Pike County [5]	Summit, MS, USA	CC	New	4	G.E.	PG7241FA	1,126.0	127.6	#DIV/0!	Turbine Air Systems	
2004	City of San Antonio	Leon Creek, TX, USA	SC	New	4	G.E.	LM 6000	193.7	37.9	#DIV/0!	Turbine Air Systems	
2004	West Minnesota Municipal	Exira Station, IA, USA	SC	New	2	G.E.	LM 6000	96.9	13.8	#DIV/0!	Turbine Air Systems	
2003	GE/Escatron	Spain		New	4	G.E.	LM 6000	168.0			Munters	
2003	Zorlu Enerji	Turkey		New	1	G.E.	LM 6000	42.0			Munters	
2003	AAF/Fars Iran	Iran		New	2	G.E.	Frame 9E	246.0			Munters	
2003	Form/Akin Tekstil	Turkey		Existing	1	Solar	Taurus 60	5.0			Munters	
2003	Tuma Turbomach/ Pakistan	Pakistan		New	1	Solar	Taurus 60	5.0			Munters	
2003	AES Sylvarena	Sylvarena, MS, USA	SC	New	3	G.E.	LM 6000	145.2	27.0	#DIV/0!	Turbine Air Systems	
2003	BTU Energy - Bryan Energy Facility	Bryan, TX, USA	SC	New	1	G.E.	LM 6000	45.0			Stellar	
2003	NRG - Calpine - Brazos Valley	Thompsons, TX, USA	SC	New	2	G.E.	PG7241FA	631.0	59.0	#DIV/0!	Turbine Air Systems	
2003	Calpine - Stony Brook	Stony Brook, NY, USA	SC	New	1	G.E.	LM 6000	48.4	9.0	#DIV/0!	Turbine Air Systems	
2003	DENA - Deming Energy Facility	Deming, NM, USA	CC	New	2	G.E.	7FA	340.0			Stellar	
2003	DENA - Fayette Energy Facility	Fayette, PA, USA	CC	New	2	G.E.	7FA	340.0			Stellar	
2003	DENA - Grays Harbor Energy Facility	Grays Harbor, WA, USA	CC	New	2	G.E.	7FA	340.0			Stellar	
2003	DENA - Hanging Rock Energy Facility	Hanging Rock, OH, USA	CC	New	4	G.E.	7FA	680.0			Stellar	
2003	DENA - Moapa Energy Facility	Apex, AZ, USA	CC	New	4	G.E.	7FA	680.0			Stellar	
2003	Glendale - Grayson	Glendale, CA, USA	SC	New	1	G.E.	LM 6000	48.4	9.0	#DIV/0!	Turbine Air Systems	

	Turbine Installation Data								Hot Weather Power Enhancement from TIC [1]		TIC System Developer, Designer, Equipment Supplier(s), or Installer	
Initial Year of TIC Operation	CT Plant Owner / Operator	CT Plant Location	Simple or Combined Cycle (SC or CC)	TIC Applied to Existing or New CTs	Quantity of CTs	CT Make	CT Model	CT Plant I.S.O. Output (MW)	TIC Power Increase (MW)	TIC Power Increase (%)	**TICA Member with Primary Involvement**	**Other TICA Member(s) Contributing Products or Services to the Project**
2002	GE/Lawrence County	Canada		New	6	G.E.	LM 6000	288.0			Munters	
2002	VAW	USA		New	6	P&W	Twin Pack	150.0			Munters	
2002	AAF/Covert	USA-MI		New	3	MHI	501G	762.0			Munters	
2002	AAF/Tupelo	USA-MS		New	2	MHI	501G	508.0			Munters	
2002	Pneumafil/Santa Cruz	Brazil		New	2	SW	501F	372.0			Munters	
2002	Pneumafil/Tractabel II	USA		New	2	SW	501G	508.0			Munters	
2002	Pneumafil/Fisk Peakers	USA		New	3	SW	501F	558.0			Munters	
2002	Pneumafil/Tractabel	USA		New	2	SW	501G	508.0			Munters	
2002	Pneumafil/Allegheny	USA		New	2	SW	501F	372.0			Munters	
2002	Pneumafil/Norte Fluminense	USA		New	3	SW	501F	558.0			Munters	
2002	GE/Ompa Ponca			New	1	G.E.	LM 6000	45.0			Munters	
2002	Stadtwerke Erfurt			Existing	2	G.E.	LM2500	50.0			Munters	
2002	Swanbank			New	1	ABB	GT 26	270.0			Munters	
2002	Pacific Corp			New	3	G.E.	LM 6000	126.0			Munters	
2002	Zorlu Enerji			Existing	2	EGT	Tempest	15.0			Munters	
2002	Nuh Enerji			Existing	1	G.E.	LM 2500	27.0			Munters	
2002	Zorlu Enerji			Existing	1	G.E.	LM 2500	27.0			Munters	
2002	Zorlu Enerji			Existing	2	G.E.	LM 6000	84.0			Munters	
2002	Oglethorpe			New	2	Siemens	V84.2	340.0			Munters	
2002	Black Hills Power - Las Vegas Cogen	Las Vegas, NV, USA	CHP		4	G.E.	LM 6000	180.0			Stellar	
2002	Calpine - Bethpage	Bethpage, NY, USA	SC	New	1	G.E.	LM 6000	48.4	9.0	#DIV/0!	Turbine Air Systems	
2002	Calpine - Creed	Suisun City, CA, USA	SC	New	1	G.E.	LM 6000	48.4	9.0	#DIV/0!	Turbine Air Systems	
2002	Calpine - Feather River	Yuba City, CA, USA	SC	New	1	G.E.	LM 6000	48.4	9.0	#DIV/0!	Turbine Air Systems	
2002	Calpine - Gilroy	Gilroy, CA, USA	SC	New	3	G.E.	LM 6000	145.2	27.0	#DIV/0!	Turbine Air Systems	
2002	Calpine - Goose Haven	Suisun City, CA, USA	SC	New	1	G.E.	LM 6000	48.4	9.0	#DIV/0!	Turbine Air Systems	
2002	Calpine - King City	King City, CA, USA	SC	New	1	G.E.	LM 6000	48.4	9.0	#DIV/0!	Turbine Air Systems	
2002	Calpine - Lambie	Suisun City, CA, USA	SC	New	1	G.E.	LM 6000	48.4	9.0	#DIV/0!	Turbine Air Systems	
2002	Calpine C-Star - Los Esteros	San Jose, CA, USA	CC	New	4	G.E.	LM 6000	193.6	36.0	#DIV/0!	Turbine Air Systems	
2002	Calpine - River View	Antioch, CA, USA	SC	New	1	G.E.	LM 6000	48.4	9.0	#DIV/0!	Turbine Air Systems	
2002	Calpine - Wolfskill	Suisun City, CA, USA	SC	New	1	G.E.	LM 6000	48.4	9.0	#DIV/0!	Turbine Air Systems	
2002	Calpine - Yuba City	Yuba City, CA, USA	SC	New	1	G.E.	LM 6000	48.4	9.0	#DIV/0!	Turbine Air Systems	
2002	DENA - Arlington Valley Energy Facility	Arlington, AZ, USA	CC	New	2	G.E.	7FA	340.0			Stellar	
2002	DENA - Hot Spring Energy Facility	Hot Spring, AR, USA	CC	New	2	G.E.	7FA	340.0			Stellar	
2002	DENA - Murray Energy Facility	Dalton, GA, USA	CC	New	4	G.E.	7FA	680.0			Stellar	
2002	DENA - Washington Energy Facility	Columbus, OH, USA	CC	New	2	G.E.	7FA	340.0			Stellar	
2002	El Paso - Corona Cogen	Corona, CA, USA	SC / CHP	Existing	1	G.E.	LM 5000	33.8	12.0	#DIV/0!	Stellar	
2002	TECO - Dell Generating Station	Dell, AR, USA	CC	New	2	G.E.	7FA	340.0			Stellar	
2002	TECO - McAdams Generating Facility	McAdams, MS, USA	CC	New	2	G.E.	7FA	340.0			Stellar	

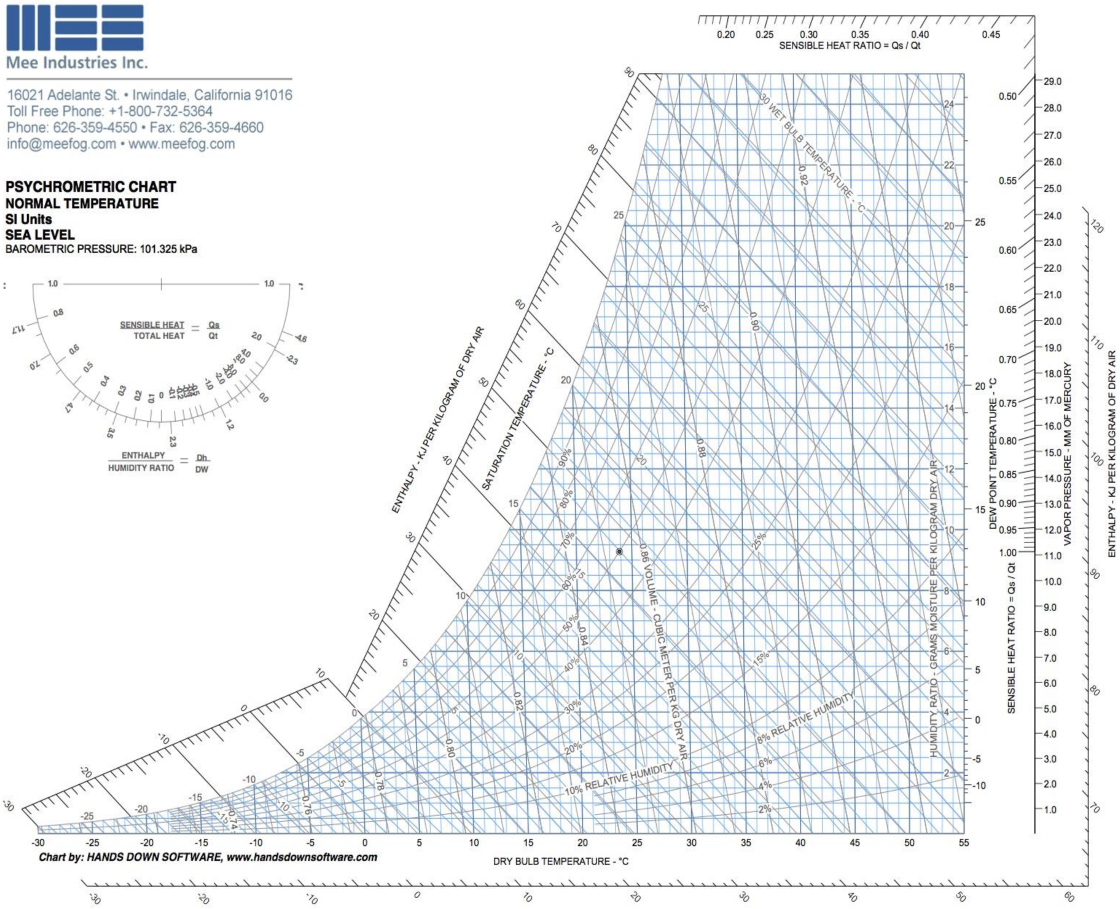
	Turbine Installation Data								Hot Weather Power Enhancement from TIC [1]		TIC System Developer, Designer, Equipment Supplier(s), or Installer	
Initial Year of TIC Operation	CT Plant Owner / Operator	CT Plant Location	Simple or Combined Cycle (SC or CC)	TIC Applied to Existing or New CTs	Quantity of CTs	CT Make	CT Model	CT Plant I.S.O. Output (MW)	TIC Power Increase (MW)	TIC Power Increase (%)	**TICA Member with Primary Involvement**	**Other TICA Member(s) Contributing Products or Services to the Project**
2001	Massac - Mid West Energy	USA-IL		New	2	G.E.	Frame 6	78.0			Munters	
2001	McGuffy	USA		New	9	P&W	Twin Pack	225.0			Munters	
2001	VAW	USA		New	14	P&W	Twin Pack	350.0			Munters	
2001	Alliance Colton	USA		New	8	G.E.	10	88.0			Munters	
2001	Pneumafil	USA-LA		New	2	SW		0.0			Munters	
2001	Pneumafil	USA		New	3	SW	501F	558.0			Munters	
2001	Pneumafil/Sithe	USA		New	2	MHI	701F	540.0			Munters	
2001	Pneumafil/Calpine Hillabee	USA		New	2	SW	501G	508.0			Munters	
2001	Pneumafil/Naconagolez	USA		New	1	SW	501G	254.0			Munters	
2001	Pneumafil/FPL RISE	USA		New	2	SW	501F	372.0			Munters	
2001	Pneumafil/Araucaria	Brazil		New	2	SW	501F	372.0			Munters	
2001	Pneumafil/Equistar	USA		New	4	SW	501F	744.0			Munters	
2001	Camfil/Farr	USA		New	2	G.E.	Frame 6B	78.0			Munters	
2001	Camfil/Farr	USA		New	3	RB	211	84.0			Munters	
2001	International Paper	USA		New	1	G.E.	Frame 6	39.0			Munters	
2001	TriGen-Cynergy	USA		New	5	Rolls Royce		70.0			Munters	
2001	Universal Silencer	USA-FL		New	1	G.E.	Frame 5 LA	23.0			Munters	
2001	Universal Silencer	USA		New	4	G.E.	10	44.0			Munters	
2001	AAF/Greystone	USA-TN		New	3	MHI	501F	558.0			Munters	
2001	AAF/Campeche	Mexico		New	1	MHI	501F	186.0			Munters	
2001	AAF/Tuxpan	Mexico		New	4	MHI	501F	744.0			Munters	
2001	AAF/Wyandotte	USA-MI		New	2	MHI	501F	372.0			Munters	
2001	AAF/Granbury	USA-TX		New	2	MHI	501G	508.0			Munters	
2001	AAF/Altamira	Mexico		New	2	MHI	501G	372.0			Munters	
2001	Bioc	Irak		New	5	ABB	GT 11 N	550.0			Munters	
2001	Covap	Switzerland		New	1	Solar	Taurus	4.0			Munters	
2001	Holden	USA		New	3	Siemens	V84.2	510.0			Munters	
2001	Monterrey	Mexico		New	4	ABB	GT 24	2,720.0			Munters	
2001	Oglethorpe	USA		New	4	Siemens	V84.2	424.0			Munters	
2001	Senoko	Singapore		New	1	ABB	GT 26	260.0			Munters	
2001	Swanbank	Australia		New	1	Alstom	GT 26	250.0			Munters	
2001	Union Carbide	USA		Existing	2	MHI	501F	340.0			Munters	
2001	DENA/PPL Global-Griffith Energy Fac	Griffith, AZ, USA	CC	New	2	G.E.	7FA	340.0			Stellar	
2001	El Paso - Macae	Macae, RJ, Brazil	SC	New	20	G.E.	LM 6000	968.0	180.0	#DIV/0!	Turbine Air Systems	
2001	Enron International - Electrobolt	Seropedica, RJ, Brazil	SC	New	6	G.E.	LM 6000	290.4	54.0	#DIV/0!	Turbine Air Systems	
2001	Enron International - Electrobolt	Seropedica, RJ, Brazil	SC	New	2	G.E.	LM 6000	96.8	18.0	#DIV/0!	Turbine Air Systems	
2001	Enron North America / Austin Energy	Austin, TX, USA	SC	New	4	G.E.	LM 6000	193.6	36.0	#DIV/0!	Turbine Air Systems	
2001	GE / Calpine - Westbrook Energy Fac	Westbrook, ME, USA	CC	New	2	G.E.	7FA	340.0			Stellar	
2001	Grays Ferry Cogeneration	Philadelphia, PA, USA	CC / CHP	Existing	1	Westinghouse	501	120.0	15.0	#DIV/0!	Cool Solutions [3]	
2001	Wildflower - Palm Springs	Palm Springs, CA, USA		New	1	G.E.	LM 6000	48.8	9.0	#DIV/0!		
2001	Wildflower - San Diego	San Diego, CA, USA	SC	New	2	G.E.	LM 6000	97.6	18.0	#DIV/0!		

	Turbine Installation Data								Hot Weather Power Enhancement from TIC [1]		TIC System Developer, Designer, Equipment Supplier(s), or Installer	
Initial Year of TIC Operation	CT Plant Owner / Operator	CT Plant Location	Simple or Combined Cycle (SC or CC)	TIC Applied to Existing or New CTs	Quantity of CTs	CT Make	CT Model	CT Plant I.S.O. Output (MW)	TIC Power Increase (MW)	TIC Power Increase (%)	**TICA Member with Primary Involvement**	**Other TICA Member(s) Contributing Products or Services to the Project**
2000	AAF	USA		New	2	ABB	GT 24	340.0			Munters	
2000	City of Vero Beach	USA		Exisiting	1	G.E.	Frame 6	70.0			Munters	
2000	Dahlmann	India		New	1	G.E.	Frame 6F	70.0			Munters	
2000	Dynegy Midwest	USA		Exisiting	4	SW	251	196.0			Munters	
2000	Formosa Plastics	USA-LA		Exisiting	2	G.E.	Frame 6B	80.0			Munters	
2000	Hermosillo	Mexico		New	1	ABB	GT 24	170.0			Munters	
2000	La Paloma	USA		New	4	ABB	GT 24	680.0			Munters	
2000	McGuffy Systems	USA		New	40	P&W		1,000.0			Munters	
2000	Peoples Calumet	USA		New	2	ABB	GT 24	340.0			Munters	
2000	Pneumafil	USA		New	17	SW	501F	3,162.0			Munters	
2000	Pneumafil	Australia		New	2	ABB	1300	330.0			Munters	
2000	Pneumafil	USA-MA		New	4	MHI	701F	1,080.0			Munters	
2000	Smurfit	Spain		New	1	FT 8		25.0			Munters	
2000	Soyland Power	USA		Exisiting	2			56.0			Munters	
2000	Trigen-St. Louis Energy	USA		Exisiting	2	Solar	60	10.0			Munters	
2000	Wolff Walsrode	Germany		Existing	1	Sulzer	3 D	6.0			Munters	
2000	EMI / Calpine - Rumford Gen Stn	Rumford, ME, USA	CC	New	1	G.E.	7FA	170.0			Stellar	GEA
2000	EMI / Calpine - Tiverton Gen Stn	Tiverton, RI, USA	CC	New	1	G.E.	7FA	170.0			Stellar	GEA
2000	Jamaica Pub. Svc. Co. - Hunts Bay	Kingston, Jamaica	CC / CHP	Existing	1	John Brown	MS5001	25.5	2.4	10%	Munters	
2000	TECO CCPS	New Church, VA, USA	SC	New	7	G.E.	LM 6000	338.8	63.0	#DIV/0!	Turbine Air Systems	
1999	City of Lubhok	USA		Existing	1	G.E.	Frame 5	25.0			Munters	
1999	Hays	USA		New	4	ABB	GT 24	680.0			Munters	
1999	Holsten Brauerei	Germany		Existing	1	Solar	Taurus	5.0			Munters	
1999	Hunt Oil	USA		Existing	2	G.E.	Frame 5	56.0			Munters	
1999	Hunt Oil	Jemen		Existing	2	G.E.	Frame 5	56.0			Munters	
1999	Jamaica Public Utility	Jamaica		Existing	1	G.E.	Frame 5	25.0			Munters	
1999	Kali und Salz	Germany		Existing	1	Solar	Taurus	5.0			Munters	
1999	L&G E	USA		New	2	ABB	GT 24	340.0			Munters	
1999	Lake Road	USA		New	3	ABB	GT 24	510.0			Munters	
1999	MAN	Germany			1	Solar	Taurus	5.0			Munters	
1999	McGuffey	USA		New	8	P&W		200.0			Munters	
1999	OHIO_PP	USA		New	2	ABB	GT 24	340.0			Munters	
1999	PPC Greece	Greece		Existing	1	TG	20	35.0			Munters	
1999	PPC Greece	Greece		Existing	1	G.E.	Frame 5	25.0			Munters	
1999	PPC Greece	Greece		Existing	2	ABB	GT 8B	108.0			Munters	
1999	Rütgers VFT	Germany		Existing	1	G.E.	Frame 8	10.0			Munters	
1999	South Texas Electricity Group	USA		Existing	1	G.E.	Frame 5	1.0			Munters	
1999	Stone Europa Carton	Germany		Existing	1	Sulzer	3 D	6.0			Munters	
1999	unknown	USA		New	4	ABB	GT 24	680.0			Munters	
1999	Wintershall Lingen	Germany		Existing	2	G.E.	LM 2500	50.0			Munters	
1999	AES - Cartagena	Cartagena, Colombia	SC	Existing	2	G.E.	LM 5000	67.6	0.0	#DIV/0!	Turbine Air Systems	
1999	Baylor University	Waco, TX, USA	SC / CHP	Existing	1			3.0				
1999	Calpine - Clear Lake	Pasadena, TX, USA	CC / CHP	Existing	3	Westinghouse	501 D5	412.0	49.0	#DIV/0!	Turbine Air Systems	Avalon Consulting, CB&I
1999	Illinova -El Paso Energy	Danville, IL, USA	SC	New	4	G.E.	LM 6000	168.4	70.4	#DIV/0!	Turbine Air Systems	

Appendix B: Global Horizontal Irradiation Map of South Africa



Appendix C: Psychrometric Chart Normal Temperature SI Units



Appendix D: Weather Conditions in Cape Town

EAST LONDON, R.S.A
CAPE TOWN, R.S.A.

EAST LONDON					CAPE TOWN			
	Temp					Temp.		
Month	°F	°C	% R.H.		Month	°F	°C	% R.H.
Oct 2015	75	23.8	58		Oct 2015	70	21.1	38
Nov	85	29.4	40		Nov	72	22.2	46
Dec	75	23.8	78		Dec	81	27.2	46
Jan 2016	83	28.3	63		Jan 2016	83	28.3	42
Feb	79	26.1	51		Feb	83	28.3	45
Mar	**91**	**32.7**	**46**		Mar	83	28.3	51

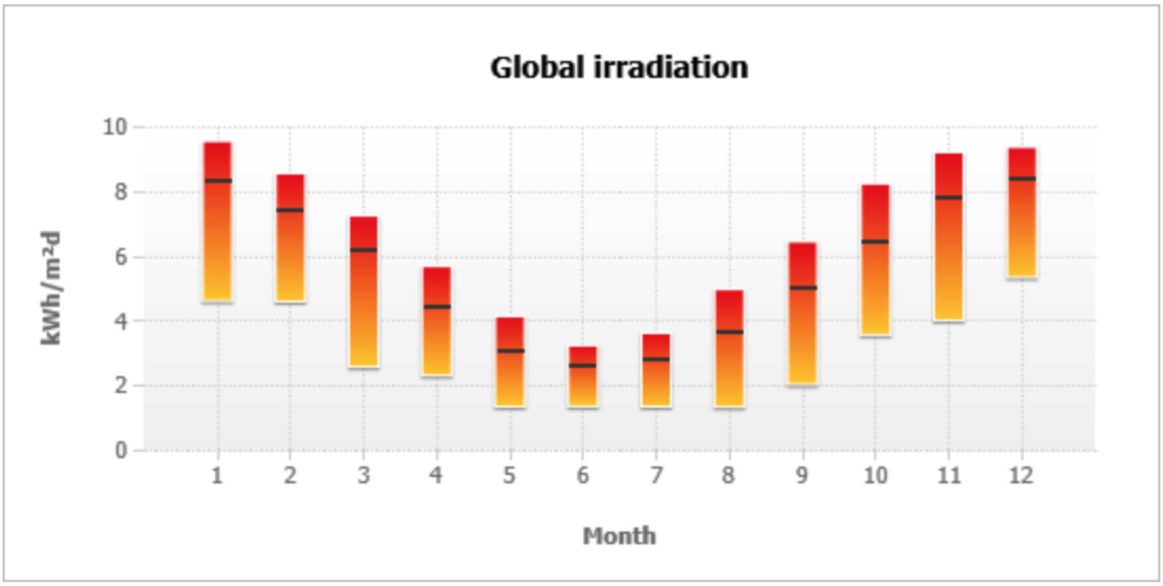
Both chosen months will require 28 grains of humidity.

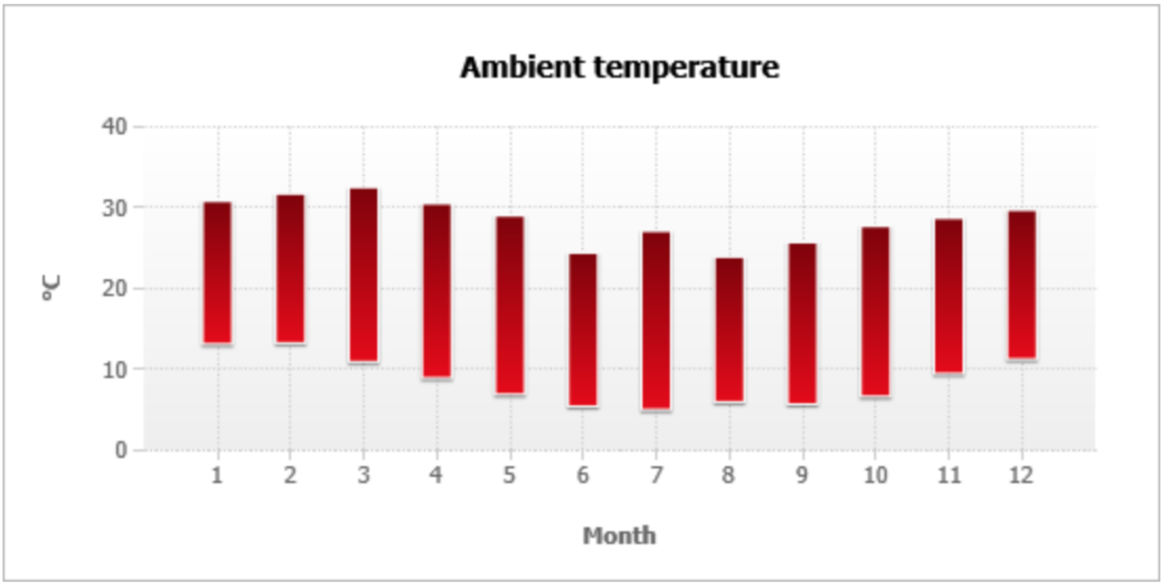
Appendix E: Meteorological Data for Cape Town

Meteorological data

The location is **Cape Town** in **South Africa (Africa)**

The annual total of global irradiation equals **1,977.68 kWh/m²a**





Appendix F: Data sheet Sunmodule Plus SW 260 ploy

Sunmodule® Plus
SW 260 poly

SOLARWORLD
REAL VALUE

PERFORMANCE UNDER STANDARD TEST CONDITIONS (STC)*

		260 Wp
Maximum power	P_{max}	260 Wp
Open circuit voltage	U_{oc}	38.4 V
Maximum power point voltage	U_{mpp}	31.4 V
Short circuit current	I_{sc}	8.94 A
Maximum power point current	I_{mpp}	8.37 A
Module efficiency	η_m	15.51 %

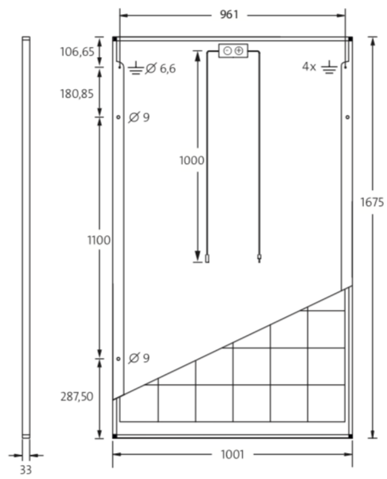
Measuring tolerance (P_{max}) traceable to TUV Rheinland: +/- 2% (TUV Power controlled)

*STC: 1000W/m², 25°C, AM 1.5

PERFORMANCE AT 800 W/m², NOCT, AM 1.5

		260 Wp
Maximum power	P_{max}	192.4 Wp
Open circuit voltage	U_{oc}	34.8 V
Maximum power point voltage	U_{mpp}	28.5 V
Short circuit current	I_{sc}	7.35 A
Maximum power point current	I_{mpp}	6.76 A

Minor reduction in efficiency under partial load conditions at 25°C: at 200 W/m², 100% (+/-2%) of the STC efficiency (1000 W/m²) is achieved.



COMPONENT MATERIALS

Cells per module	60
Cell type	Poly crystalline
Cell dimensions	156 mm x 156 mm
Front	Tempered safety glass (EN 12150)
Back	film, white
Frame	Clear anodized aluminum
J-Box	IP65
Connector	H4

DIMENSIONS / WEIGHT

Length	1675 mm
Width	1001 mm
Height	33 mm
Weight	18.0 kg

THERMAL CHARACTERISTICS

NOCT	46 °C
TK I_{sc}	0.051 %/K
TK U_{oc}	-0.31 %/K
TK P_{mpp}	-0.41 %/K

PARAMETERS FOR OPTIMAL SYSTEM INTEGRATION

Power sorting	-0 Wp / +5 Wp
Maximum system voltage SC II	1000 V
Maximum reverse current	25 A
Load / dynamic load	5.4 / 2.4 kN/m²
Number of bypass diodes	3
Operating range	-40°C bis +85°C

ORDERING INFORMATION

Order number	*Description*
82000008	Sunmodule Plus SW 260 poly

SolarWorld AG reserves the right to make specification changes without notice.
This data sheet complies with the requirements of EN 50380.

2016-03-08 EN