Thermodynamic Study of Advanced Gas Turbine Cooling System

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Submitted by:

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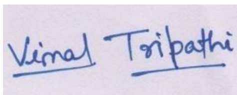
I, Vimal Tripathi, 2K18/THE/15 student of M.Tech (Thermal Engineering), hereby declare that the project Dissertation title “**Thermodynamic Study of Advanced Gas Turbine Cooling System**” which is submitted by me to the Department of Mechanical Engineering, Delhi Technological University, Delhi in partial fulfillment of the requirement for the award of the degree of Master of Technology under the supervision of **Prof. B. B. Arora**, is original and not copied from any source without proper citation. This work has not previously formed the basis for the award of any Degree, Diploma Associateship, Fellowship, or other similar title or recognition.



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I hereby certify that the Project Dissertation titled "**Thermodynamic Study of Advanced Gas Turbine Cooling System**" which is submitted by Vimal Tripathi, 2K18/THE/15, Mechanical Engineering Department, Delhi Technological University, Delhi in partial fulfillment of the requirement for the award of the degree of Master of Technology is a record of the project work carried out by the students under my supervision. To the best of my knowledge this work has not been submitted in part or full for any Degree or Diploma to this University or elsewhere.

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ABSTRACT

A gas turbine's power output is directly proportional to and limited by the mass flow rate of compressed air available to it from the air compressor that supplies high-pressure air to the gas turbine system's combustion chamber. An air compressor has a set capacity for handling a volumetric flow rate of air. Even though a compressor's volumetric capacity is constant, with changes in ambient air temperature, the mass flow rate of air it supplies to the gas turbine changes. With rising ambient temperature, this mass flow rate of air decreases as the density of air decreases as the temperature of the air increases. Therefore, when there is a rise in ambient temperature, the power output of a combustion turbine decreases below its rated capacity, so this paper deals with various inlet air cooling systems integrated into gas turbines. The purpose of this analysis is to combine two separate cooling systems (evaporative cooling system and mechanical cooling system). Modeling of thermodynamics is carried out and presented along with results that demonstrate the effect of the integrated cooling system on different gas turbine output parameters and their main environmental benefits and disadvantages. The incorporation of the inlet air cooling system reduces the air compression work required and increases the gas turbine's net working performance and thus increases the gas turbine's overall efficiency. The simulation is also carried out using various parameters to change.

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LIST OF SYMBOLS, ABBREVIATIONS

Abbreviations

GT	Gas turbine
CPR	Cycle pressure ratio
CIT	Compressor inlet temperature
SAR	Steam to air ratio
HRSG	Heat recovery steam generator
STIG	The steam recovery heat generator
TIT	Turbine inlet temperature
LHV	Lower heating calorific value
IAC	Inlet air cooling
IEC	Inlet evaporative cooling
SI	Steam injected
FC	Film cooling
VCIAC	Vapor compression inlet air cooling

Subscripts

a	Air
b	Blade
e	Exit
g	Gas
f	Fuel
i	Inlet
s	Steam
vap	Vapor
amb	Ambient
com	Compressor
evc	Evaporative cooling
cl	Coolant
gt	Gas turbine
cc	Combustion chamber
aw	Adiabatic

pt	Polytropic
DB	Dry bulb
WB	Wet-bulb
l	Latent heat
se	Sensible heat

SYMBOL

c	Blade cord
c_P	Specific heat at constant pressure (kJ/kgK)
f	The ratio of the mass flow rate of fuel to the mass flow rate of air
h	Enthalpy (kJ/kgK)
$\dot{m}$	Mass flow rate
s	Blade spacing
v	Specific volume (m^3/kg)
w	Work done
A	Area (m^2)
C	Velocity (m/s)
E	Evaporative cooler effectiveness
L	Latent heat of evaporation (kJ/kg)
M	Molar mass
P	Pressure (kPa)
T	Temperature (K)
Q_l	Latent cooling load
Q_s	Sensible cooling load
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
St	Stanton number
α	Coolant flow discharge angle
φ	Relative humidity
λ	Ratio of cooled blade surface to hot gas flow area
ε	Film cooling effectiveness

η	Efficiency
ω	Humidity

CHAPTER 1

INTRODUCTION

INTRODUCTION OF COOLING SYSTEM

The energy performs a very significant factor in any country's development, so one of the most critical issues for any nation is to satisfy its energy demand. With time, the consumption of energy is growing with the population. The target for producing electricity for traditional energy sources in India in 2016-17 was 1178 BU (billion units), which boosted to 1229.4 BU in 2017-18, 2019-20 was 1250.784 BU in 2019-20 and it will increases to 1330 Billion Unit (BU) in 2020-21 [1]. In order to meet this growing energy demand, either the number of power plants to be expanded or the turbine inlet air cooling system can be used and will minimize the pressure of rising electricity demand to some degree and the cost of building an inlet air cooling system is low relative to building a new power plant.

The gross work output of the gas turbine relies significantly on the ambient air temperature, i.e. the compressor entry stage, since the work performed by the compressor relies specifically on the ambient air temperature to be compressed, meaning that when the ambient air temperature is higher, the compressor absorbs more power and the net power decreases. By cooling the inlet air, net power output is increased and performance improved. In hot and dry regions where the ambient temperature is high, these inlet air cooling systems are more efficient.

Gas turbines are often used during on-site power cycles to satisfy the high demand. Unfortunately, this time generally coincides with the day's highest temperatures, which significantly influence the gas turbine performance; this turbine output loss due to higher ambient air temperature can be balanced by using inlet air cooling.

Inlet air cooling cools the ambient air until it reaches the gas turbine compressor, which provides high-pressure & high-temperature air into the combustion chamber from which high-pressure hot air enters the combustion turbine. Inlet air cooling is sometimes known by different names, including inlet air cooling in the combustion turbine, inlet air cooling in the compressor, air cooling in the combustion turbine, and gas turbine inlet air cooling.

Inlet air cooling systems are not just the ideal option for a high demand period (the total cost is around 1/3rd of a new gas turbine's capital investment), but they also increase the gas turbine heat output. The Inlet air cooling system improves the power per unit of fuel by increasing the heat output, thereby making the gas turbine more productive and lowering carbon emissions.

PRINCIPLE

A gas turbine's power output is directly related to the mass of the compressed air flow rate from the air compressor that delivers high-pressure air to the gas turbine system's combustion chamber. Both air compressors have a set capacity to accommodate volumetric air pressure at a given RPM. While the volumetric carrying capacity of all compressors is constant, with increases in ambient air temperature the mass flow rate of air that it supplies to the gas turbine varies. This air mass flow rate declines as atmospheric temperature increases because the air density declines as air temperature rises. Consequently, a gas turbine's output power falls below its rated potential under ISO conditions (15°C, RH (relative humidity) 60%, and absolute pressure (sea level) 101.325kPa) with atmospheric temperature rises above 15°C. The inlet air cooling system allows air density to be improved by reducing the temperature and thereby helps to increase the mass flow rate of air into the gas turbine and to boost the gas turbine performance.

BENEFITS OF INLET AIR COOLING SYSTEM

- Overall Gas turbine power production rises.
- The auxiliary part is often used to cool.
- Improve heat output from the turbine.
- Air Temperature constant at the inlet.
- Avoid the fluctuation of gas turbine output due to atmospheric conditions.
- Lower price per produced MW compared with the new turbine gas turbine.
- Acknowledge green technology with carbon credits.

TYPE OF INLET AIR COOLING SYSTEM

➢ Water evaporation system

  - Wetted media evaporative cooling
  - Fogging
  - Wet compression/over-spraying

- Heat transfer system
    - Absorption chiller cooling
    - Mechanical refrigerative cooling
    - Thermal energy storage

WETTED MEDIA EVAPORATIVE COOLING

As the title implies, evaporative cooling is achieved with the help of evaporation in wetted media. This machine includes an air cleaner, wetted media, water storage tank, and a water pump. First ambient air is permitted to flow via air filters where all unwanted particles are removed after filtered air is allowed to travel via wetted media in which water enters from one side and filtered air from the other side, wetted media is a wet porous surface, there are several forms of wetted media system and honeycomb is best between them. For hot and dry areas, this form of device is fine.

In wetted media, as wet surface and hot air interacted so some water from the wet surface take heat from hot air and evaporate and lower the temperature of the hot air inlet, in this method sensible heat loss from inlet air functions as a latent heat gain for water and this entire process is an isenthalpic process along with cooling and humidification process as a psychometric process.

The salt content of water rises due to the continuous evaporation of water from wetted media, which raises the risk of scaling such that a purifying system is needed and there is also a continuous supply of water the mass of water needed to compensate for the evaporated water depends on the degree of cooling necessary, the atmospheric conditions and the flow rate of the turbine gas. The only downside to this system is that the temperature of the hot inlet air below the temperature of the wet-bulb equivalent to the atmospheric state will not be decreased by this system.

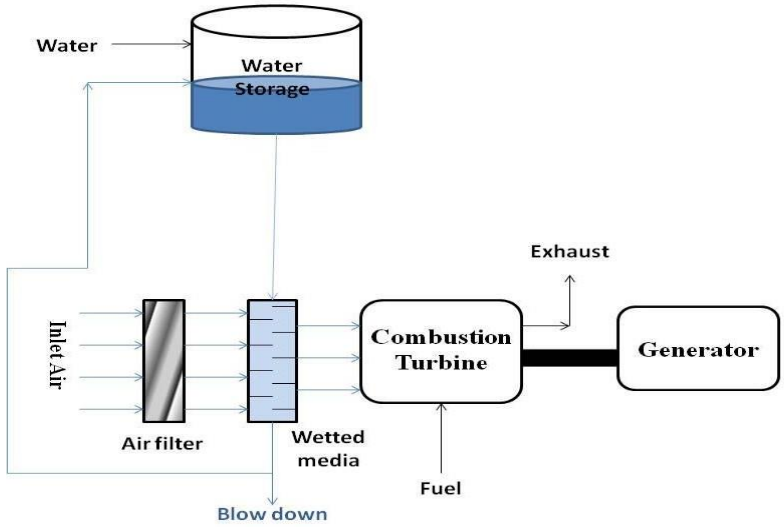

Figure 1.1 Turbine inlet air cooling system using wetted evaporative system [2]

FOGGING

In this method, the theory of cooling is the evaporation of water and the atomization of water with the aid of the nozzles. This machine consists of a de-mineralized water plant, a de-mineralized water storage tank, an inlet air filter, a fogging spray (array of nozzles), and a gas turbine.

Initially, water is de-mineralized to decrease the risk of compressor blade fouling; normally mist remover is added after fogging spray to eliminate evaporated water droplets that can damage the compressor blade. As fogging, as small water droplets and hot air come into contact, then water droplets extract heat out of hot air and evaporate and lower the temperature. Inlet hot air, in this process, responsive heat loss from inlet air functions as a latent heat benefit for water, and this whole process is isenthalpic along with the process of cooling and humidification as a psychometric method.

Developing a fogging device by bearing in mind that the volume of the droplets should be close to 20microns to maximize the evaporation rate and the duct length between fogging spray and compressor is such that the time taken by the water droplet to cover this distance should be greater than the time needed to evaporate and the evaporation time depends on the diameter of the droplet, DBT, WBT. This method has some drawbacks, such as its high reliance on relative humidity, required de-mineral water, and the minimum temperature obtained is the humid bulb temperature corresponding to the ambient state.

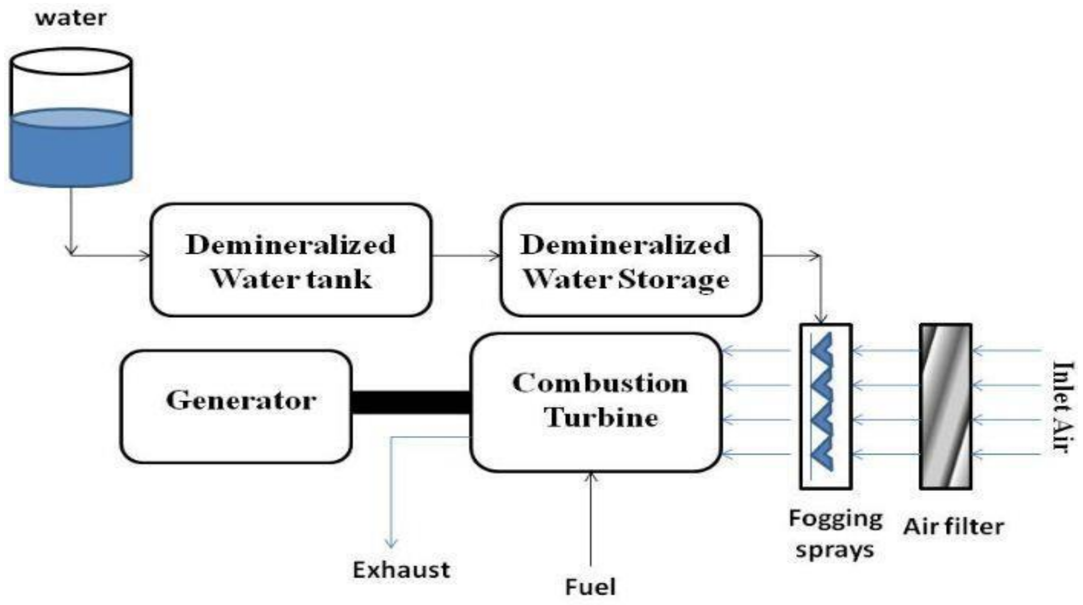

Figure 1.2 Turbine inlet air cooling using fogging. [2]

WET COMPRESSION / OVERSPRAY

In wet compression, the water droplet is applied as a fog to the inlet air much as it was used for fogging. The amount of fog applied, however, is a lot more than can be evaporated under the atmospheric air conditions. The inlet air stream takes the extra water into the gas turbine compressor parts from which it further evaporates, cools the compressed air, and produces more mass with the aid of evaporative cooling technology to improve the gas turbine performance beyond the conceivable. The quantity of excess mist supplied to the compressor relies on where the fog is applied to the inlet portion of the gas turbine system. This is called continuous cooling since continuous cooling during compression is often caused by evaporation of extra fog.

Compared to fogging, there are compressor inlet areas in damp compression nozzles where more water droplets can reach the compressor. During compression temperature changes in compressed air in each stage and with compressed temperature water-holding capability also changes so that in subsequent stages additional water droplets are absorbed. Net gas turbine output increases in wet compression, such that this method is often used in commercial and military aircraft to achieve additional thrust during takeoff.

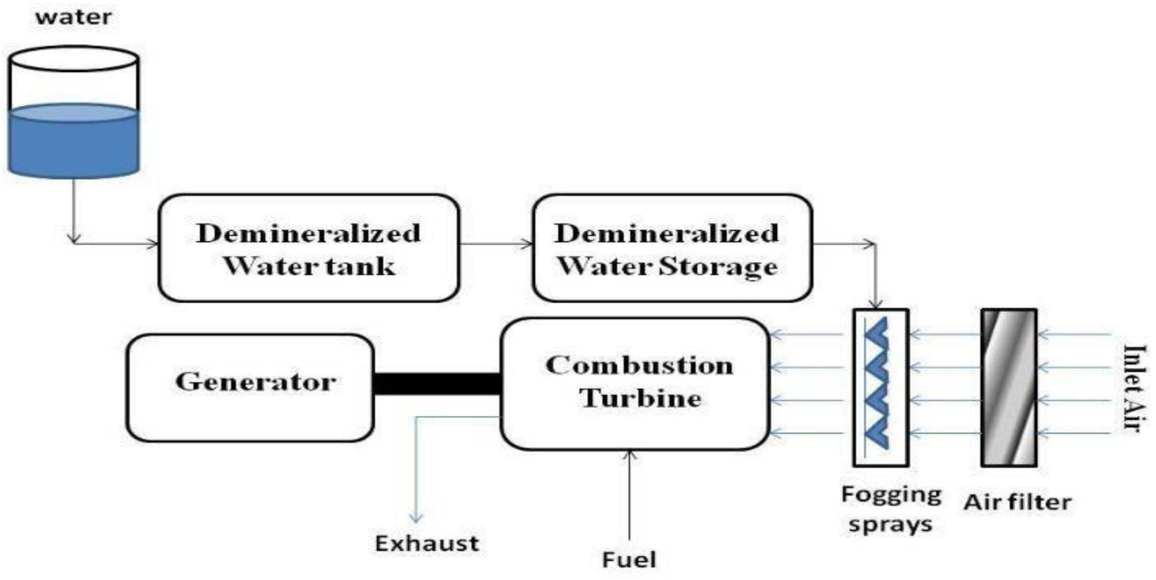

Figure 1.3 Turbine inlet air cooling with wet compression/overspray [2]

ABSORPTION CHILLING COOLING

Instead of using energy, absorption chillers use heat as the energy source. This supply of heat energy is often waste steam from a combined cycle or hot water from a cogeneration facility, and the cooling mechanism can be powered. Absorption chillers have a very low coefficient of performance (COP) relative to mechanical chillers, but it can be taken into account that these cooling chillers usually use excess heat, which lowers running costs and is a bonus for this device.

In absorption chilling cooling, cooling is achieved by vapour absorption method in which mostly lithium bromide-water 0r water-ammonia pair is used in each pair one refrigerant role and another one absorbent role. This device functions on low-grade energy or wasted energy such as energy in turbine exhaust, solar radiation, and another benefit of using this device is that the minimum temperature is not limited by atmospheric conditions such as in the evaporative cooling system, however, the minimum temperature that can be achieved is restricted by ice crystal formation at the compressor inlet, ice crystal formation badly harmed the compressor blade.

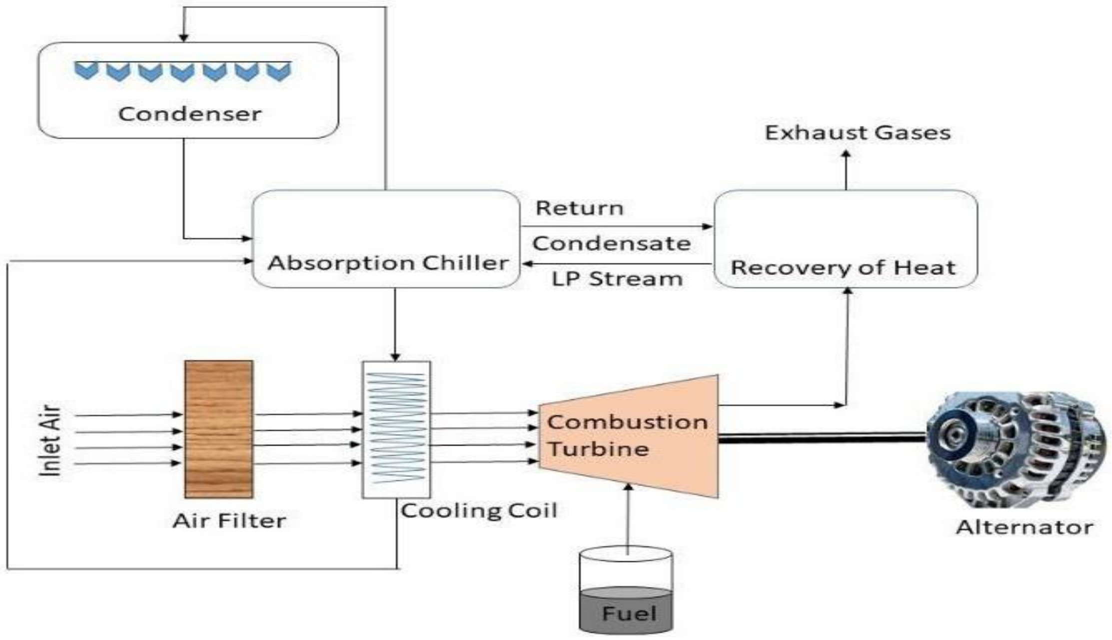

Figure 1.4 Turbine inlet air cooling using absorption chiller [2]

MECHANICAL REFRIGERATIVE COOLING

Mechanical cooling relying on a VCRS cooling loop in which water is cooled down to low temperature and poured into a cooling coil mounted within the turbine's filter house. In the cooling coil, cooled water collects the heat present in the air and cools it up to the optimum temperature.

As opposed to an evaporative cooling system, one of the key advantages of an inlet air chilling system is that the chilling system is not confined to cooling the air to the predominant temperature of the wet filament. The TIAC system's ability to cool the air just below the WBT results in a substantially improved power output. In hot, humid climates, where evaporative cooling is very inefficient, this is of special importance. TIAC devices for mechanical vapour compression are also capable of dry operation with no usage of water. Evaporative cooling may be potentially feasible in such settings, but the vast amounts of needed freshwater are clearly not always sufficient.

The vapour compression refrigeration system (VCRS) is used in the mechanical refrigeration cooling system to cool the inlet air. The cooling capacity of such a device is estimated in the cooling ton(TR) and 1 TR capacity means 12000 Btu/hr heat removal. The compressor is a large power consumption unit in VCRS that can be powered by electricity, natural gas, turbine steam. The process of this cooling system is the same as the basic VCRS loop, the only difference being the inlet air in the evaporator is cooled here.

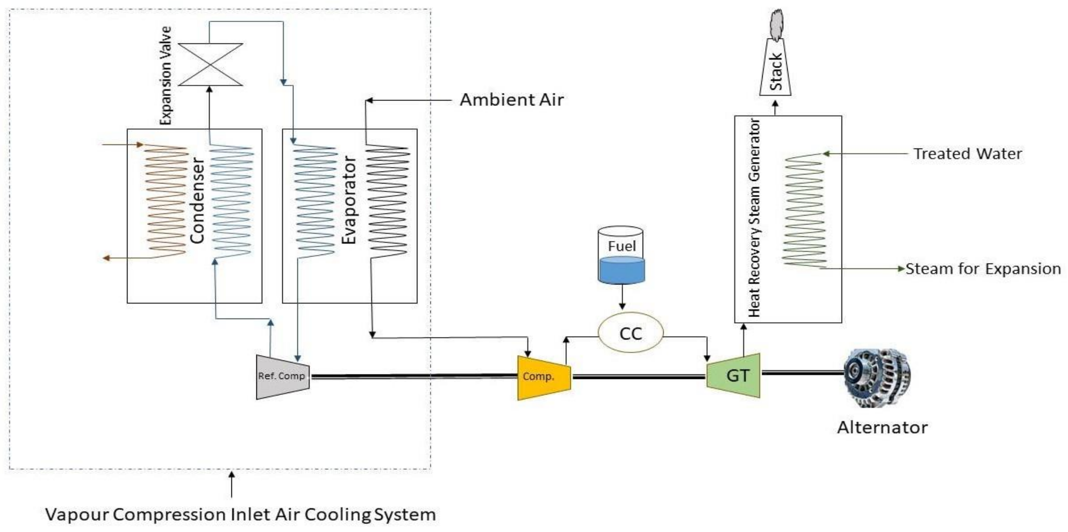

Figure 1.5 Turbine inlet air cooling system using mechanical refrigerative cooling.[2]

THERMAL ENERGY STORAGE

Compared to other heat transfer systems, the turbine inlet cooling system with thermal energy storage uses an external component that is a storage tank (vapour compression inlet cooling system, inlet cooling system for vapour absorption). In the thermal energy storage inlet air cooling system, the cooling system is decoupled to cool inlet air during the off-time of peak load and used to cool liquid in case of sensitive thermal energy storage for heat or to form ice in case of latent thermal energy storage for heat and to use heat during peak load from these thermal energy storage tanks.

TES Tank is a thermal collector that allows the storage during the off-peak period of chilled water or ice made. During on-peak time, this energy is later used to cool the inlet air from the turbine and increase its power output. A TES tank decreases the power and operating costs of the refrigerant facility, providing chilled water when the demand is very low, which typically correlates with the night when the ambient temperature is low and the chillers work well. The usage of surplus power generation would take place throughout the day.

Thermal Energy Storage (TES) operates in two phases: charging and discharge. The colder and less concentrated returning water floats on top of the accumulated chilled water during the discharge process. The storage water is delivered and withdrawn at low velocity, in basically horizontal flow, so that inertial effects are dominated by the buoyancy forces. When the stratified storage tank is filled, chilled supply water enters at the base of the tank through the diffuser and returns water leaves through the diffuser at the top of the tank to the chiller device.

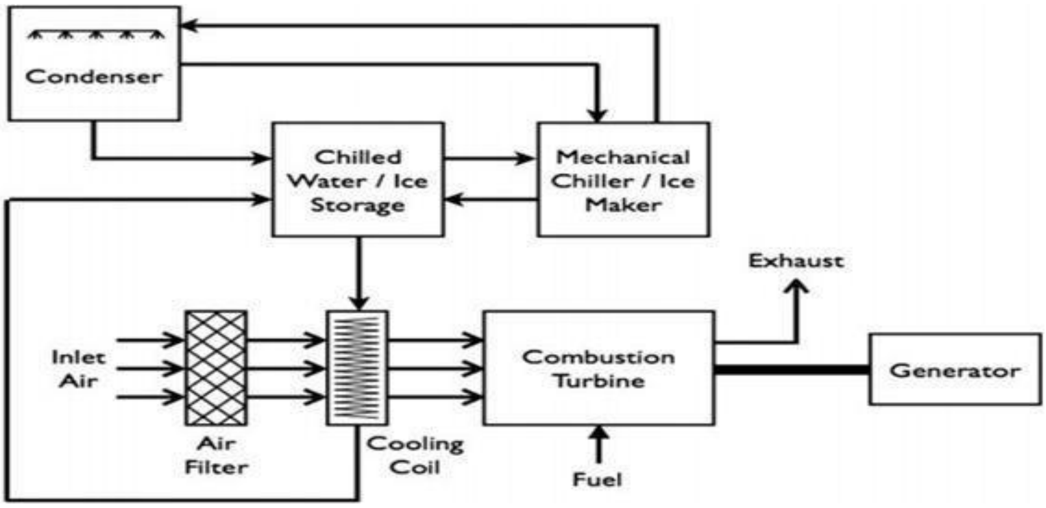

Figure 1.6 Turbine inlet air cooling system using energy storage tank [2]

1.3 COMPARATIVE STUDIES

Table 1

Benefits and drawback of various technologies, adapted from Al-Ibrahim[2]

Technology	Benefits	Drawback
Evaporative Cooling		
Stream of hot ambient air comes in directs contact with water flowing in evaporative cooler and drop temperature of hot ambient air from its dry bulb temperature to wet bulb temperature corresponding to ambient condition.	• Very low capital and installation cost. • Simple in design and easy to handle no expert requires to operate • No limitation on time and duration of inlet air-cooling operation • Low parasitic power consumption • Low operational cost • Quickly come in operation.	• Limitation on achieving minimum temperature • Minimum achievable temperature is constraint by ambient condition • Purified water required in large amount. • Due to direct contact with water chance of rusting and scaling increase which increase mentaiance cost • Limited capacity improvement

High-Pressure Fogging

Excess water spray on inlet air in the form of fog to achieve continuous cooling and cool inlet to wet bulb temperature.	• Low initial cost. • Excess water spray on inlet air, reduce its temperature and compressor work which further increase net work output. • There is no limitation time to use this system. • Annual maintenance cost is low • Consumption of parasitic power is minimum • Deliver cool air quickly and constantly	• Minimum temperature attains is limited by ambient condition. • Water requirement is more than evaporative cooling • Required water free of salts • Number of filter required is more • Limited capacity improvement

Absorption Cooling

This system use waste heat or low grade energy for reducing the temperature of inlet air to desired temperature.	• Minimum attain temperature is not restricted by ambient condition • Best way to utilize waste energy from turbine exhaust. • Inlet air cooling	• Initial cost is high • More operational cost • Any Minimum temperature can achieve • A team of expert is required to

	operation is not limited by time • Parasitic power losses is less. • Performance is better than evaporative or fogging system.	maintain and operate the system. • This system is not good for open cycle gas turbine • Required large amount of waste heat and cooling tower with other system • System takes time to provide low temperature air.
Mechanical Refrigeration A simple vapour compression refrigerative system is used to cool incoming air.	• Minimum attainable temperature is not constraint by ambient condition • Any minimum desirable temperature is achieve • No limitation on time or duration of inlet air-cooling operation • Simple and reliable design relatively	• High initial cost • During peak time required high amount electrical power • High operational cost • Expert required for operation • More installation time • Another chilled water cooling system required • Parasitic load is more compare to

	• Perform good compare to evaporative and fogging	evaporative and fogging
Ice Thermal Energy Storage Decouples the cooling system during off peak load and use cooling system to form ice which can be used during peak load	• Any desirable minimum temperature can achieve • During peak time electric power demand is low • At night cooling system used to form ice which can be used in peak load time • Compare to evaporative and fogging performance is good	• High initial cost • System is more complex • Large storage space is required to store ice • High level of expert required to operate • Delivery time is more.

CHAPTER 2

LITERATURE REVIEW

In the present time demand for electricity is increasing day by day so to meet that increasing demand for electricity it is necessary to increase the efficiency of the present power plant. The efficiency of the power plant can be easily increased by using a turbine inlet air cooling system so nowadays a lot of research work is going on this topic.

The first time gas turbine inlet air cooling system used was a direct air conditioning system for a power plant in Battle Creek, Michigan (USA) in 1987-88.

M.Abdulrahman et al.[2] deeply study In 2010 the different inlet air cooling systems more specific to Saudi Arabia and compare different inlet air cooling systems depending on their benefits and drawbacks.

MacCracken [3] gives in 1992 a brief history and overview of the progress and the potential of thermal storage in off-peak turbine inlet cooling and it was the second time when the turbine inlet air cooling system used.

B.J.Kitchen et al.[4] examine In 1995 the combustion turbine for capacitive enhancement by using an inlet air cooling system and develop a new inlet air cooling effectiveness factor.

Maurizio De Lucia et al.[5] study In 1996 the benefits of using different compressor inlet air cooling systems for the cogeneration plant. Absorption and evaporative cooling systems are considered and their performance and economic benefits compared for the dry low-NOx LM6000 version and conclude that an integration of the 2 systems proved to be able to give both maximum performance enhancement and net economic benefit.

William E. Stewart [6] publish In 1999 a book with the name of Design guide Combustion turbine inlet air cooling system which talks about the size of the duct, type of nozzle to get fine droplets, and type of refrigerant used.

C.B. Meher-Homji et al.[7] did a deep study In 2000 on inlet air cooling by fogging and got results that cover practical consideration related to placing of nozzles, type and quality of water used, and operational and maintenance issues.

Kim Hoon Kyounh et al. [8] study In 2011 the evaporation of water fogging system with the help of four different heat and mass transfer modes(diffusion, natural convection, strokes convection, perturbed strokes) and find the critical water injection ratio dividing the low and high fogging cases.

Sanjay et al.[9] thoroughly study In 2014 the evaporative cooling and vapor compression cooling together along air film cooling were adopted for gas turbine blades to get maximum network output.

Carmona Jose [10] In 2014 concluded the different results related to evaporative cooling specific for Lagos-Nigeria related to the maximum cost of cooling, operating and maintenance cost

Giovanna et al.[11] did a techno-economic study In 2015 for gas turbine inlet air cooling for combined cycle power plant for different climate condition and used cold water thermal storage as a cooling method and conclude that operational hours and power output augmentation were higher in hotter climates and wet climate required huge thermal storage thus increasing investment cost.

Anoop Kumar Shukla et al.[12] publish a Research paper In 2016 on the performance of gas turbines with inlet evaporative cooling and steam injected in the combustion chamber and this paperwork as base paper for our study and our study use all equation which is used in this paper.

Ahmed Ali Abdel Rahman et al.[13] work on inlet air cooling with the help of solar energy and observe that the peak demand for electricity coincides with the peak of solar radiation. And concluded that the absorption chiller system is best to work with solar energy and also show that plants without inlet air cooling system produce more carbon emission compare to plant using inlet air cooling system.

CHAPTER 3

SYSTEM DESCRIPTION

INTRODUCTION

In this chapter, the layout of two different systems is discussed individually which are taken from the research work of Anoop Kumar Shukla on evaporative cooling [12] and Achintya Sharma on VCRS cooling[14] which also work as base paper for an integrated system which is discussed in details as a proposed system.

ASSUMPTION TAKEN FOR THERMODYNAMICS MODELING OF SYSTEM.

- The analysis is done assuming a steady flow condition.
- Air is taken as a working fluid which considers as an ideal gas.
- The compressor used in the study is an axial flow compressor.
- Specific heats of the working fluid are the only function of temperature.
- Evaporation in evaporative cooling assuming to be adiabatic in nature.
- Assuming an adiabatic combustion process.
- Assuming complete combustion of fuel in the combustion chamber.

EVAPORATIVE COOLING SYSTEM.

A schematic diagram of steam injected gas turbine using an evaporative inlet air cooling system shown in fig3.1 First atmospheric air is sucked by the evaporative cooling system and in evaporative cooling system temperature of hot ambient airdrop from dry bulb temperature to wet bulb temperature corresponding to ambient condition and allow to entre in compressor where it compresses to high temperature and pressure and entre in the combustion chamber where the combustion of fuel is taking place.

In the combustion chamber along with compressed air and fuel, some steam from HRSG is also injected to enhance the mass flow rate of flue gases and some steam is used to cooled the gas turbine blade so that the gas turbine work on some higher temperature. Atmospheric gas which is cool by evaporative cooler is considered as ideal gas and fuel used in the combustion chamber is natural gas whose composition is given in table 2. The pressure loss in the combustion chamber and coolant injection in the turbine blade is taken as given in Table2. Mathematical modeling of components of the steam injected gas turbine-based power plant is carried out on the basis of mass and energy balance.

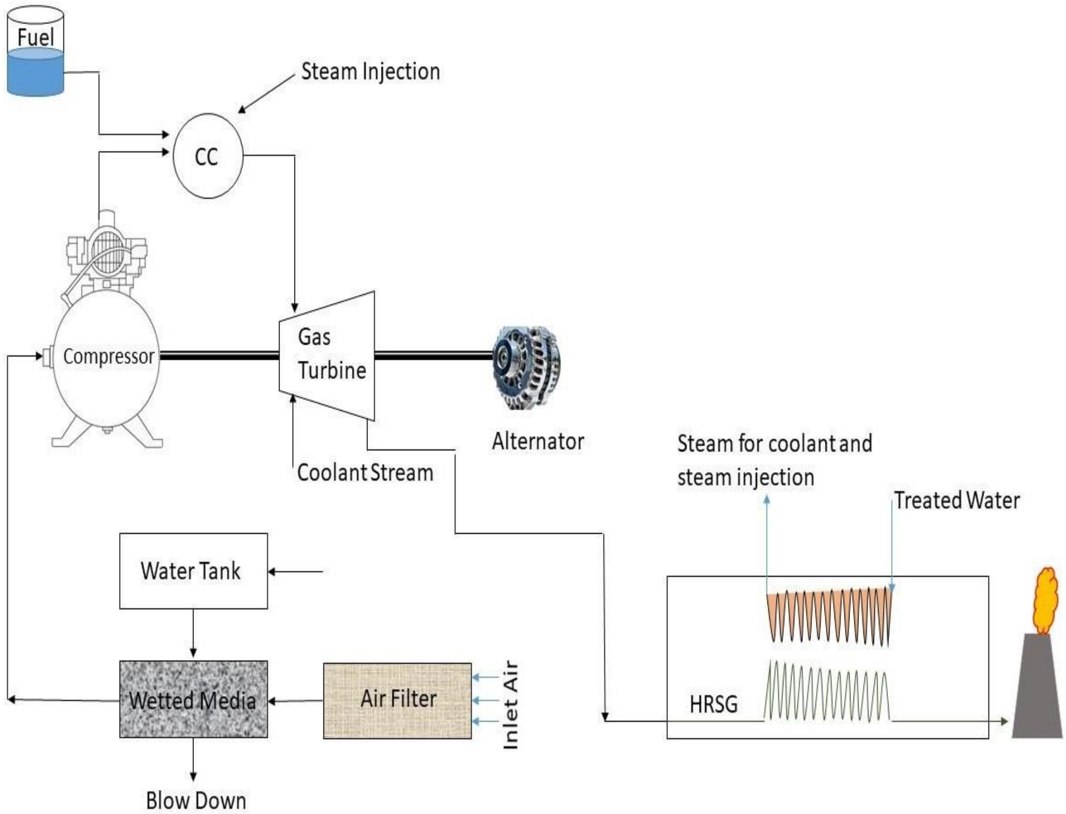

Figure 3.1 Block diagram of steam injected gas turbine with evaporative inlet air cooling.

Compressor

In this analysis, an axial flow compressor is used and polytropic efficiency is used to consider all the losses occurring in the compressor. Temperature and pressure at any stage of the compressor are related by the following relation

$$\frac{dT}{T} = \left(\frac{R_{com}}{\eta_{pt.com} * C_{p.com}}\right)\frac{dP}{P} \quad (3.1)$$

Where $\eta_{pt.com}$ is the polytropic efficiency of the compressor, $C_{p.com}$ is the specific heat at constant pressure and R_{com} is the gas constant in kJ/kgK.

And

$$= C_{p.c} - C_{v.com} \tag{3.2}$$

Where

$$C_{p.} = C_{p.} - \omega_{a.} * C_{p.vap} \tag{3.3}$$

$$C_{v.} = C_{v.a} - \omega_{a.i} * C_{v.vap} \tag{3.4}$$

$C_{p.a}$ and $C_{v.a}$ are specific heats at constant pressure and at constant volume of air respectively, in kJ/kgK and $\omega_{a.i}$ is specific humidity of air at the compressor inlet. $C_{p.a}$ and $C_{v.a}$ are the only functions of mean temperature along with the compressor which calculated as

$$C_{p.a} = 1.048 - 3.837 * 10^{-4} T_m + 9.453 * 10^{-7} T^2_{\overline{m}} \\ 5.491 * 10^{-10} T^3_m + 7.929 * 10^{-14} T^4_m \tag{3.5}$$

$$C_{v.a} = C_{p.a} - 0.287 \tag{3.6}$$

Where T_m is the mean temperature across the compressor

$C_{p.vap}$ and $C_{v.vap}$ are specific heat at constant pressure and specific heat at constant volume of water vapour accordingly and are the only function of average temperature along with the compressor.

$$C_{p.vap} = 1.8778 - 5.112 * 10^{-4} T_m + 1.9157 * 10^{-6} T^2_{\overline{m}} \\ 1.367 * 10^{-9} T^3_m + 3.723 * 10^{-14} T^4_m \tag{3.7}$$

$$C_{v.vap} = C_{p.vap} - 0.4614 \quad (3.8)$$

Energy and mass balance across compressor give compressor work done as

$$\dot{W} = \dot{m}_{com.e} * h_{com.e} - \dot{m}_{com.i} * h_{com.i} \quad (3.9)$$

Estimation of cooling load.

The sensible cooling load (Q_s) of atmospheric air coming into the evaporative cooler is calculated with the help of the following equation.

$$Q_s = \frac{V_{amb.a}}{v_{wair}} C_{p\,amb} (T_{DB\,amb} - T_{com.i}) \quad (3.10)$$

where, $V_{amb.a}$, $T_{DB\ amb}$, $T_{com.i}$ are volumetric flow rate, ambient dry bulb temperature, and compressor inlet temperature respectively and v_{wair} is the specified volume of wetted air per kg of dry air which is got from the given equation

$$v_{wair} = (0.287 + \omega_{amb,DB} * 0.462) \frac{T}{P_{atm}} \quad (3.11)$$

where $\omega_{amb,DB}$ is specific humidity of ambient air which is calculated with the help of saturation pressure of ambient air(P_{sat}), relative humidity(φ), the molar mass of water vapour(M_v), and molar mass of air(M_a) as shown in given equation:

$$\omega_{amb,DB} = \frac{P_{sat}}{(\frac{P_{sat}}{\varphi} - P_{sat})} \frac{M_v}{M_a} \quad (3.12)$$

Latent heat cooling load can be calculated from the following equation if the compressor entry temperature is less than or equal to dew point temperature of atmospheric air:

$$Q_l = \frac{V_{amb.a}}{v_{wair}} \{\omega_{amb,DB} (C_{p,v} T_{DB,amb} + L) - \omega_{c,i} (C_{p,v} T_{com,i} + L) -$$

$$(\omega_{amb,DB} - \omega_{c,i}) C_{p,,}\} \quad (3.13)$$

Where C_p, L, ω_c, and C_p, are specific heat of water vapour at constant pressure, latent heat of evaporation of water at 273.15 K, specific humidity at compressor entry temperature, and specific heat of liquid water at constant pressure respectively.

Evaporative cooling

In evaporative cooling, cooling is based on the fact that evaporation causes a cooling effect so when hot ambient air comes in contact with flowing water then water takes latent heat of vaporization from hot air and cool the hot ambient air. Dry bulb temperature of air exiting the evaporative cooler is calculated as:

$$E = \frac{T_{DB,i} - T_{DB,o}}{T_{DB,i} - T_{WB,o}} \tag{3.14}$$

where E is the effectiveness of evaporative cooler and DB, WB denotes dry bulb and wet bulb respectively and i , o use in subscript show inlet and outlet of evaporative cooler.

The evaporative process is assuming to be adiabatic and the temperature of the water being evaporated is equal to T_a, after evaporation specific humidity of air at the exit of the evaporator is given by:

$$\omega_{a,o} = \frac{C_{p,a}(T_{DB} - T_{a,o}) + \omega_{amb,DB}(C_{p,v}T_{DB} + L - C_{p,w}T_{a,o})}{C_{p,v}T_{a,o} + L - C_{p,w}T_{a,o}} \tag{3.15}$$

Mass balance across evaporator gives the mass flow rate of water required for evaporation

$$\dot{m}_{w,evc} = \frac{V_{amb.}}{v_{w,air}}(\omega_{a,o} - \omega_{DB}) \tag{3.16}$$

Combustion chamber

Heat is added in the combustion chamber when fuel (natural gas) is burning, in this combustion chamber mixture of compressed air from the compressor and steam from HRSG burn with fuel. Assuming ab adiabatic steady flow combustion process with given combustion efficiency.

From simple mass and energy balance following equation are obtain:

$$\dot{m}_{com.} + \dot{m} + \dot{m}_{s.cc} = \dot{m}_g \tag{3.17}$$

where $\dot{m}_{com.e}$ is the mass flow rate at compressor exit, $\dot{m}_f$ mass flow rate of fuel, $\dot{m}_{s.cc}$ the mass flow rate of steam injected and $\dot{m}_g$ mass flow rate of flue gas.

$$\dot{m}_{com,} = \dot{m}_a(1 + \omega_{com,e}) \tag{3.18}$$

Energy balance

$$\dot{m}_a * h_{a,.e} + \omega_{com,e} * h_{\omega,com,e} + \dot{m}_f * LHV * \eta_{CC} + \dot{m}_{s,cc} * h_{s,cc}$$

$$= (\dot{m}(1 + \omega_{com,e}) + \dot{m}_f + \dot{m}_{s,cc}) * h_{cc,e} \tag{3.19}$$

Fuel air ratio

$$f = \frac{\dot{m}_f}{\dot{m}_a} = \frac{(1+\omega_{cc.e}+SAR)_{,e} - SAR*h_{s,cc} - (h_{a,com,e}+\omega_{com,e}h_{\omega,com,e})}{LHV*\eta_{CC} - C_{pg}T_{cc,}} \tag{3.20}$$

where SAR is steam to air ratio

$$SAR = \frac{\dot{m}_{s,}}{\dot{m}_a} \tag{3.21}$$

Cooled gas turbine blade

Gas turbine blade cooling is used to increase the working temperature of the gas turbine and improve the creep resistance, fatigue resistance, and working lifetime. Here gas turbine blade is cooled by film cooling in which steam is allowed to flow over the turbine surface from one or more locations where higher is achieve which is the leading edge of the turbine blade and some steam also injected into the downstream of the blade. By using the turbine blade cooling working temperature of the gas turbine increase but it increases the cost of the system. A flat plate surface model is used to model, the film cooling, in which the coolant requirement for gas turbine blade can be given as:

$$\frac{\dot{m}_{cl}}{\dot{m}_g} = \lambda * St_g * \frac{C_{pg}}{C_{p.cl}}\left\{\frac{(T_{gt,i} - T_b) - \varepsilon_{aw}(T_{gt,i} - (T_{cl,i} + \eta_c(T_b - T_{cl,i})))}{\eta_c(T_b - T_{cl,i})}\right\} \tag{3.22}$$

where λ is the ratio of internal cooled blade surface area to external hot gas flow area (A_b/A_g) and η_c is the cooling efficiency.

$$= 2c/s * \cos \propto \tag{3.23}$$

Where ∝ is the coolant flow discharge angle, c is the blade chord and s is the blade spacing

Adiabatic film cooling effectiveness (ε_{aw}) is given as:

$$\varepsilon_{aw} = \frac{T_g - T_{aw}}{T_g - T_{cl,e}} \tag{3.24}$$

Average Stanton number (St_g)

$$St_g = \frac{Nu_g}{Re_g Pr_g} \tag{3.25}$$

where Nu_g is Nusselt number, Re_g is Reynolds number, and Pr_g Prandtl number

$$Nu_g = 0.037 * Re_g^{0.8} Pr_g^{1/3} \tag{3.26}$$

Gas turbine

Turbine work is done ($\dot{W}_{gt}$)

$$\dot{W} = \dot{m}_{gt,i}(h_{gt,i} - h_{gt,e}) + \dot{m}_{cl}(h_{cl,i} - h_{cl,e}) \tag{3.27}$$

Net specificwork output(w_{net}) $\quad w_{net} = w_{gt} - w_{com}$ (3.28)

Thermal efficiency(η_{th}) $\quad \eta_{th} = \frac{w_{net}}{\dot{m}_f * LCV}$ (3.29)

Specific fuel consumption(sfc) $\quad sfc = \frac{3600 * \dot{m}_f}{w_{net}}$ (3.30)

Table 2

Input parameters used in evaporative cooling system analysis.

Parameter	Symbol	Unit
Compressor	(a) Mechanical efficiency ($\eta_{m.com}$) = 98.5	%
	(b) Polytropic efficiency ($\eta_{pt.com}$) = 92	%
Gas turbine	(a) Exhaust pressure = 0.109	MPa
	(b) Turbine blade temp. (Tb) = 1123	K
	(c) Polytropic efficiency ($\eta_{pt.gt}$) = 92	%
	(d) Exhaust hood temperature loss = 5	K
Evaporative cooling	(a) Evapurative cooler effectiveness (E) = 90	%
Combustion chamber	(a) Lower heating calorifc value of fuel (LHV) = 44,769	kJ/kg
	(b) Pressure loss (Δp) = 3% entry pressure	%
	(c) Combustion efficiency (η_{cc}) = 99.5	%
	(d) Fuel composition CH_4 = 90%, C_2H_6 =4.5%, CO_2 = 4%, and N_2 = 1.5% by weight	%
	(e) Air composition O_2 = 21%, N_2 = 79% by volume	%
	(f) Steam injection pressure = 3	MPa
	(g) Steam injection temperature = 610	K
Gas properties	(a) Specific heat at constant pressure $C_P = f(T)$	kJ/kg K
	(b) Enthalpy $h = \int C_P \, dT$	kJ/kg K
Film cooling	(a) Film cooling efficiency (η_c) = 70	%
	(b) Lamda (λ) = 10	
	(c) Prandtl number (Pr_g) = 0.7	
	(d) Reynolds number (Re_g)=$1*10^6$	
	(e) Adiabatic wall effectiveness (ε_{aw}) = 40	%

System Validation

Consider research paper by Anoop Kumar Shukla and Onkar Singh on "Performance evaluation on steam injected gas turbine based on power plant with inlet evaporative cooling" as base paper for studying the evaporative cooling system for a gas turbine. A program is made on EES software which considers all the equations and assumptions used in base paper and results are computed based on a program that is compared with the results of the base paper to validate the program.

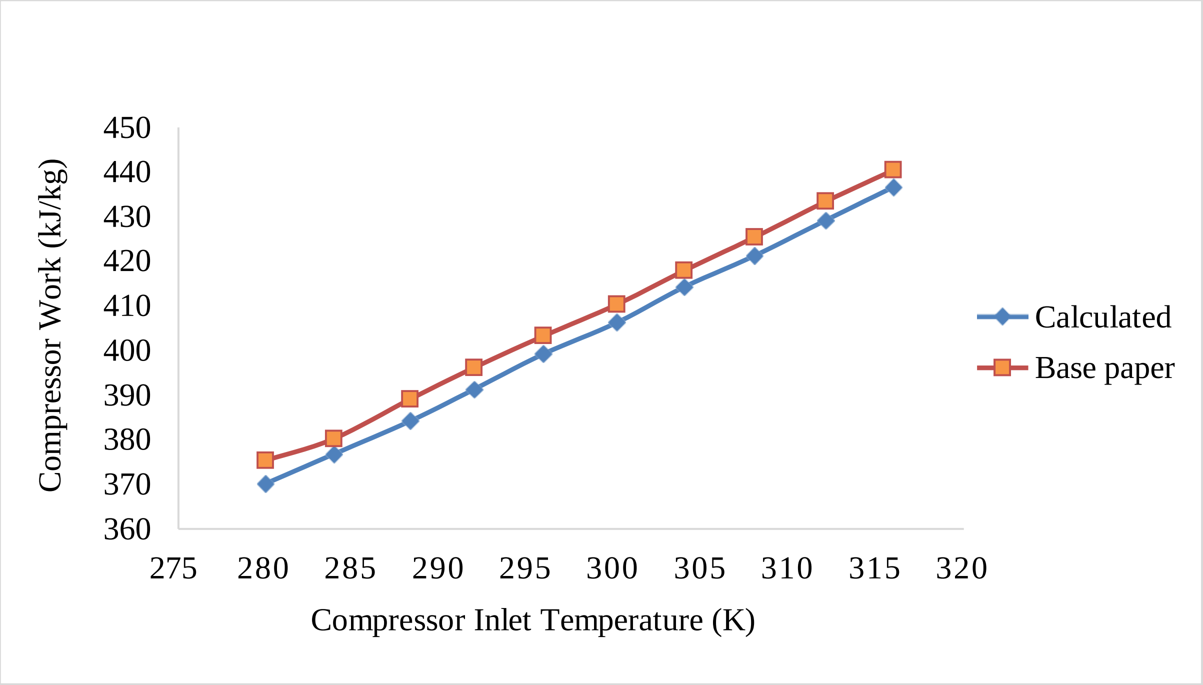

Figure 3.2 Variation of compressor work with compressor inlet temperature for system validation.

From fig 3.2 it can be depicted that compressor work increase with an increase in compressor inlet temperature for both the cases. At a lower CIT of 282K percentage error in the compressor, work is 1.2% and at a higher CIT of 318K percentage error in the compressor, work drop to 0.65%.

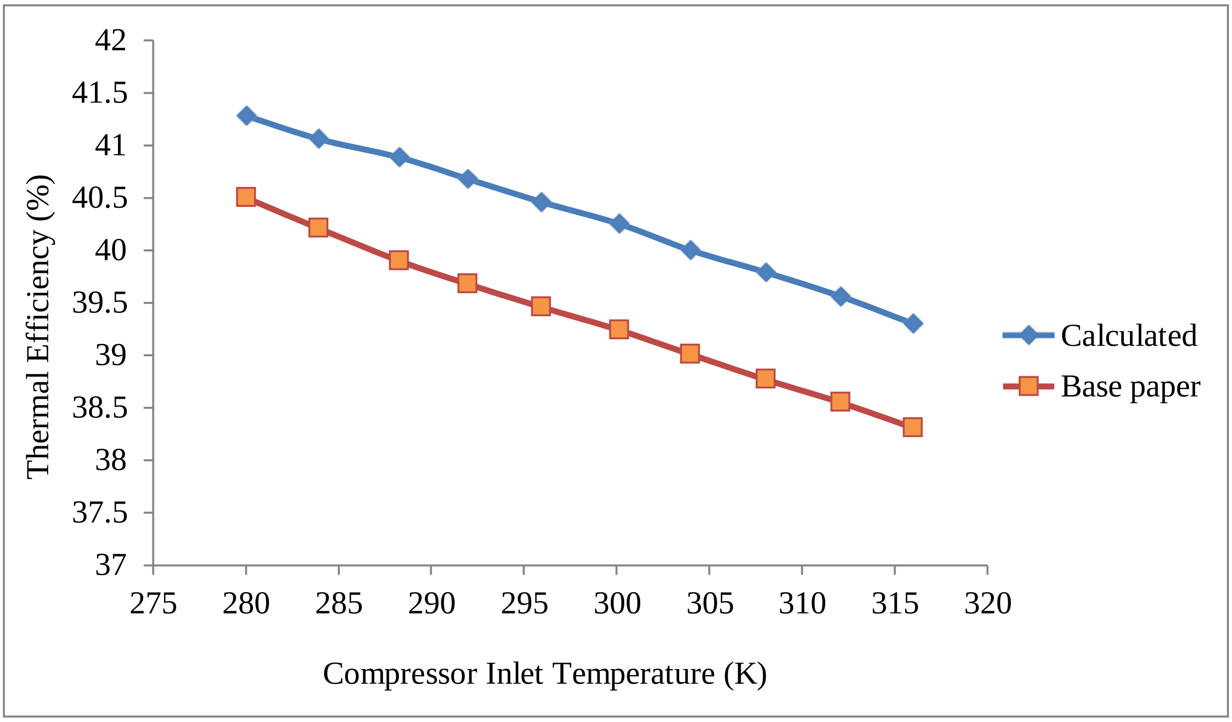

Figure 3.3 Variation of Thermal Efficiency with Compressor inlet Temperature for system validation.

Fig 3.3 depicts that thermal efficiency decreases with an increase in CIT for both cases. At a lower CIT of 282K percentage error in thermal efficiency is 1.9% and at a higher CIT of 318K percentage error in thermal efficiency increase to 2.6%.

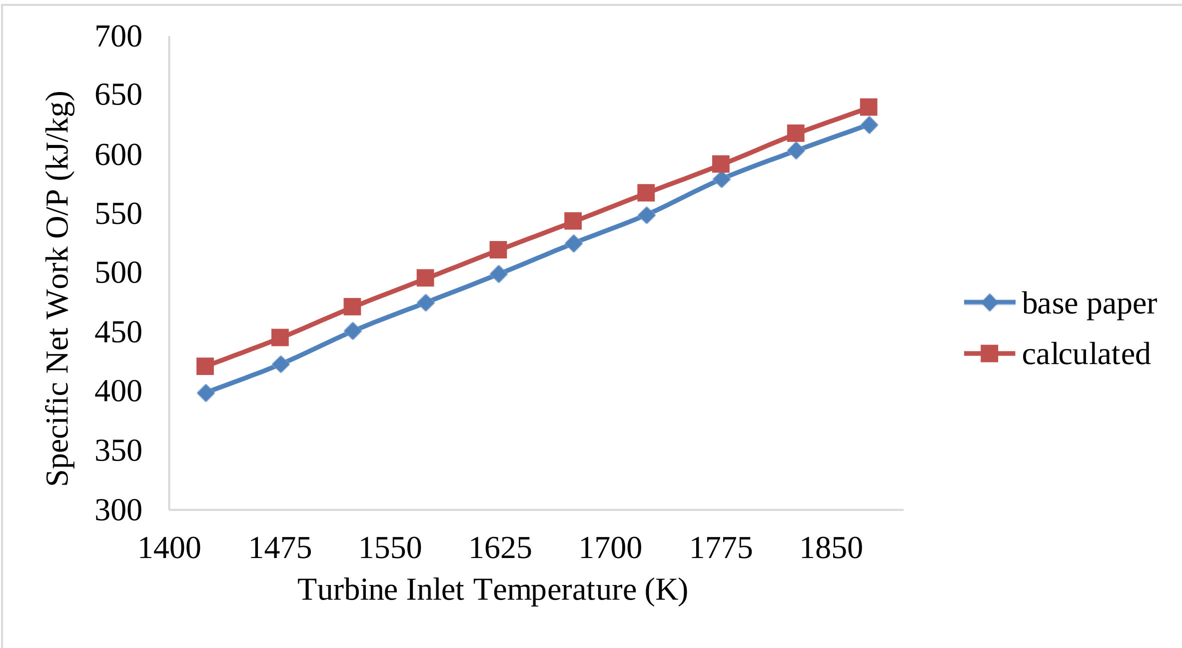

Figure 3.4 Variation of specific net work output with TIT for system validation.

From fig 3.4 it can be seen that net-specific work O/P increase with an increase in TIT. At lower TIT of 1450K percentage error is 4.75% and at higher TIT of 1900K percentage error is 2.3%.

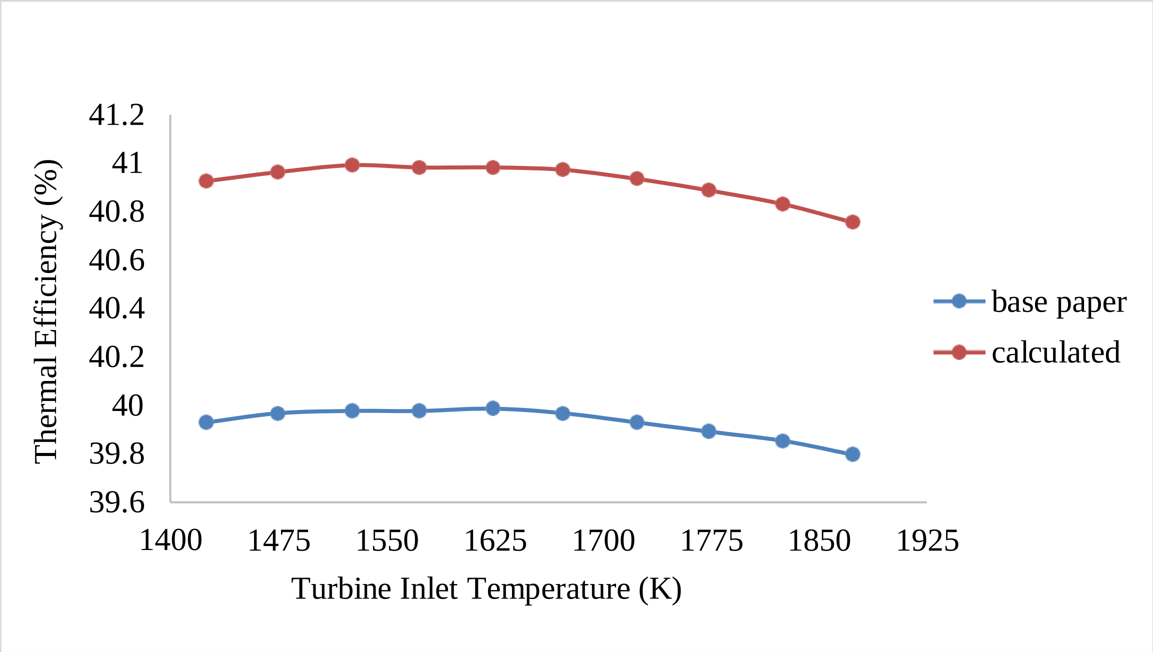

Figure 3.5 Variation of Thermal Efficiency with TIT for system validation.

Fig 3.5 depicts the decrease in thermal efficiency with an increase in turbine inlet temperature. At lower TIT of 1450K percentage error in thermal efficiency is 2.5% and at higher TIT of 1900 percentage in thermal efficiency is 2.5%.

From all the graphs used for system validation, it is observed that the maximum percentage error is 4.5% which is less than 5% so it can be said that the program is giving satisfactory results which are closed to base paper results.

VCRS COOLING SYSTEM

As shown in figure 3.6, in VCRS cooling system first ambient air come in VCRS unit from where it’s cooled to the desired temperature and then allow to entre in compressor where it compressor to more pressure after that compressed air come in the combustion chamber where it burns along fuel and allowed to expand in the turbine.

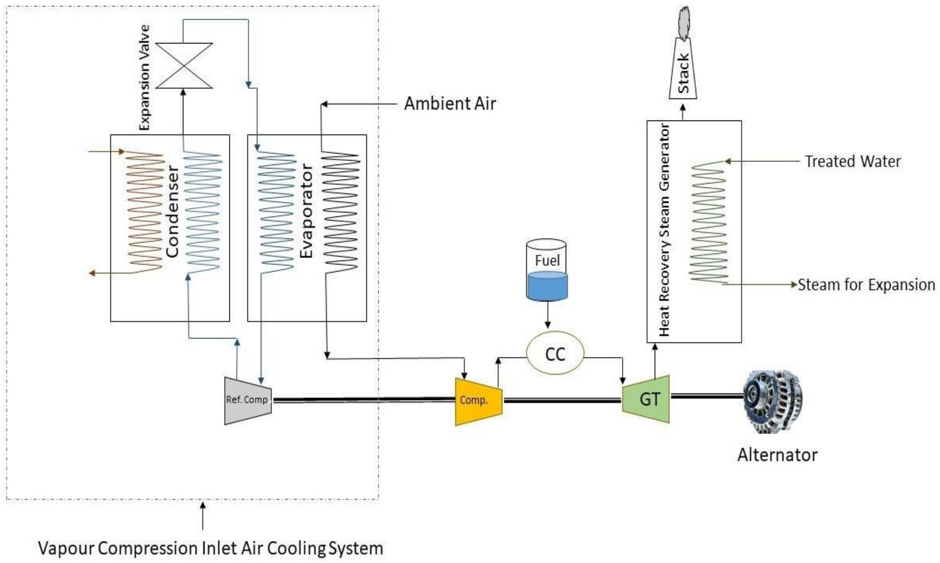

Figure 3.6 Block diagram of a gas turbine with a mechanical inlet air cooling system.

Vapor Compression Inlet Air Cooling.

Vapor compression inlet air cooling system is work on the conventional vapor compression refrigeration system which is also used in air conditions and the fridge main power-consuming device in this system is the compressor . Hence, the actual refrigerating compression work is given as:

$$\dot{W}_{rcom} = \frac{Q_{net,}}{\eta_m \eta_{el} \eta_{vol}} = \frac{Q_{net,}}{\eta_{eu}} \tag{3.31}$$

where η_{m}, η_{el} and η_{vol} are mechanical efficiency and electrical efficiency and volumetric efficiency respectively. Refrigeration work for a given value of COP is calculated with the help of the following equation:

$$W_{ref} = \frac{Q_{net,}}{COP_{vc}\eta(1-\mu * x)^n} \tag{3.32}$$

Here μ is an empirical constant that changes with the types of refrigerants and x are the quality of the refrigerant at the exit of expansion valve system. R22 is taken as a refrigerant for the present cooling system for which μ is 0.69. The exponent n depends on the number of compression and expansion stages. In this, the value of n is equal to one for a simple refrigeration cycle with one stage compressor.

The total cooling load is the addition of latent cooling load and sensible cooling load.

$$Q_{net,} = Q_s + Q_l \tag{3.33}$$

$$Q_s = \frac{V_{amb.a}}{v_{wair}} C_{p\,amb} (T_{DB\,amb} - T_{com.i}) \tag{3.34}$$

$$Q_l = \frac{V_{amb.a}}{v_{wair}} \{\omega_{amb,DB} (C_{p,v} T_{DB,amb} + L) - \omega_{c,i} (C_{p,v} T_{com,i} + L) - (\omega_{amb,DB} - \omega_{c,i}) C_{p,,}\} \tag{3.35}$$

Compressor

Following temperature-pressure relation is used

$$\frac{dT}{T} = (\frac{R_{com}}{\eta_{pt.com} * C_{p.com}}) \frac{dP}{P} \tag{3.36}$$

Where $\eta_{pt.com}$ is the polytropic efficiency of the compressor, $C_{p.com}$ is the specific heat at constant pressure and R_{com} is the gas constant in kJ/kgK.

Combustion Chamber

Fuel air ratio after mass and energy balance

$$f = \frac{\dot{m}_f}{\dot{m}_a} = \frac{(1+\omega_{cc.e})_{,e} - (h_{a,com,e} + \omega_{com,e} h_{\omega,com,e})}{LHV * \eta_{CC} - C_{pg} T_{cc,}} \tag{3.37}$$

Gas Turbine

Considering the polytropic expansion in the gas turbine following equation is used to get the relation between pressure & temperature at any stage of the expansion process in the gas turbine

$$\frac{dT}{T} = \left(\frac{dP}{P}\right)^{\eta\left(\frac{\gamma-1}{\gamma}\right)} \quad (3.38)$$

Net specificwork output(w_{net}) $$w_{net} = w_{gt} - w_{com} - w_{ref} \quad (3.39)$$

Thermal efficiency(η_{th}) $$\eta_{th} = \frac{w_{net}}{\dot{m}_{f} * LCV} \quad (3.40)$$

Table 3

The input parameter uses VCRS cooling system analysis.

Parameter	**Symbol**	**Unit**
Gas Turbine	(a) Temperature of turbine blade (Tb) = 1073	K
	(b) Polytropic efficiency ($\eta_{pt,gt}$) = 92	%
	(c) Exhaust pressure = 0.109	MPa
Combustion Chamber	(a) Pressure loss = 3% of entry pressure	%
	(b) Lower calorific value of fuel (LCV) = 44769	kJ/kg
	(c) Combustion Efficiency (η_{cc}) = 99.5	%
Compressor	(a) Mechanical efficiency ($\eta_{m,com}$) = 98.5	%
	(b) Polytropic efficiency ($\eta_{pt,com}$) = 92	%
Vapour Compression Inlet air cooling	(a) Refrigerant R22	R22
	(b) Energy use efficiency = 85	%
	(c) Refrigeration Efficiency =45	%

System Validation

Consider research paper by Anoop Kumar Shukla, Achintya Sharma, Meeta Sharma, and Shivam Mishra on “Performance Improvement of Simple Gas Turbine Cycle with Vapor Compression Inlet Air Cooling” as base paper for studying the mechanical cooling system for a gas turbine. A program is made on EES software which considers all the equations and assumptions used in the base paper and results are computed based on a program which is compared with the results of the base paper to validate the program.

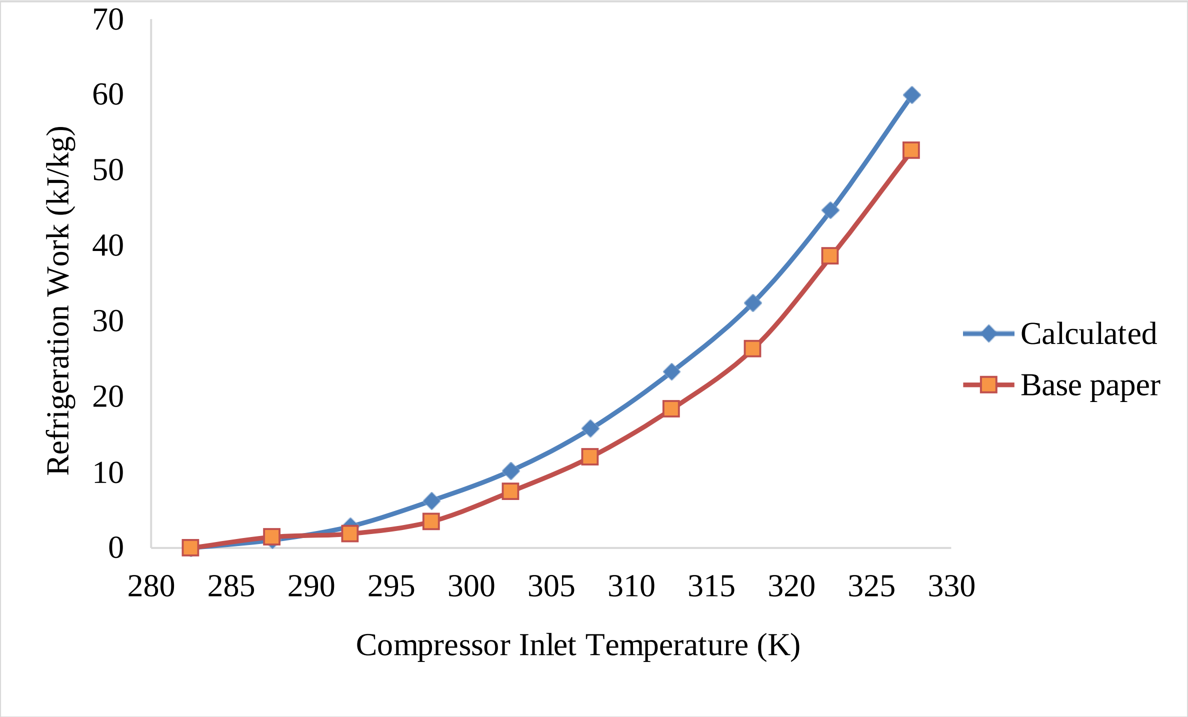

Figure 3.7 Variation of Refrigeration Work with Ambient Temperature for system validation.

From fig 3.7 it can be depicted that refrigeration work is increase with an increase in ambient temperature. At a lower ambient temperature of 285K, there is very little deviation and considerable variation is observed after 295K which increases with an increase in ambient temperature, at a higher ambient temperature of 330K percentage error is 1.4%.

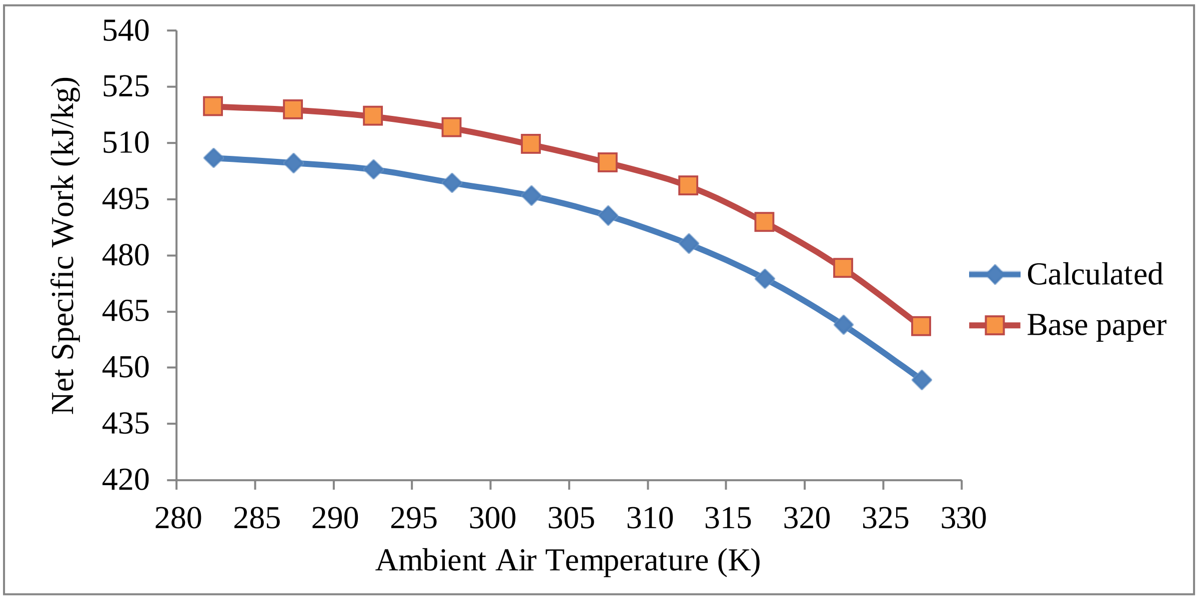

Figure 3.8 Variation of Net Specific Work with Ambient Temperature.

At a lower ambient temperature of 285K percentage error is 2.7% and at a higher ambient temperature of 330K percentage error increase to 3.1%.

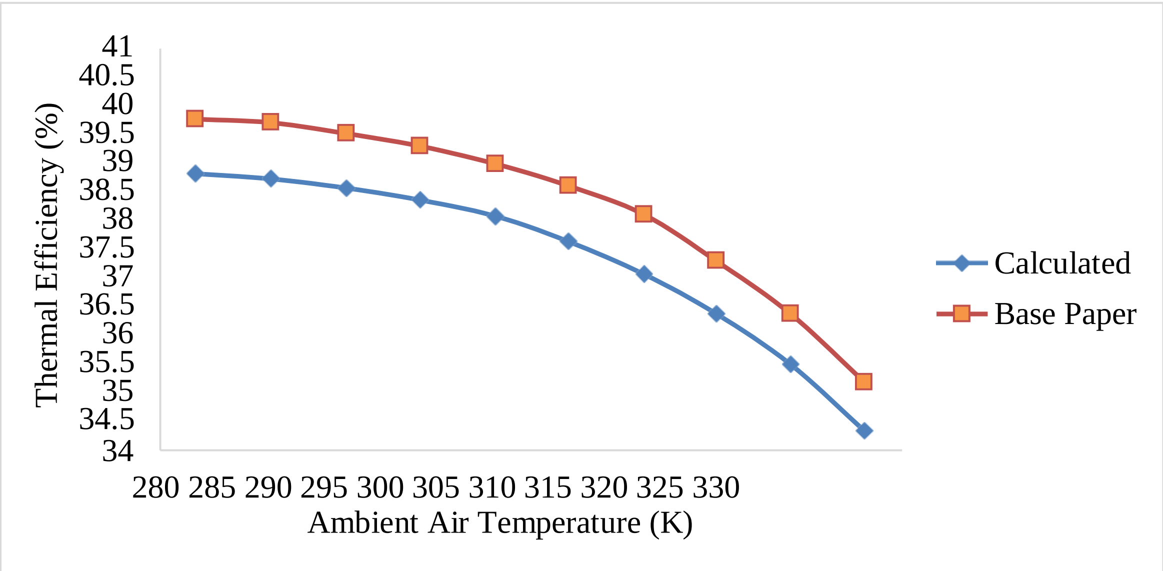

Figure 3.9 Variation of Thermal Efficiency with Ambient Temperature.

At a lower ambient temperature of 285K percentage error in thermal efficiency is 2.4% and at a higher ambient temperature of 330K percentage, the error is 2.5%.

From all the graphs used for system validation, it is observed that the maximum percentage error is 4.5% which is less than 5% so it can be said that the program is giving satisfactory results which are closed to base paper results.

INTEGRATED COOLING SYSTEM

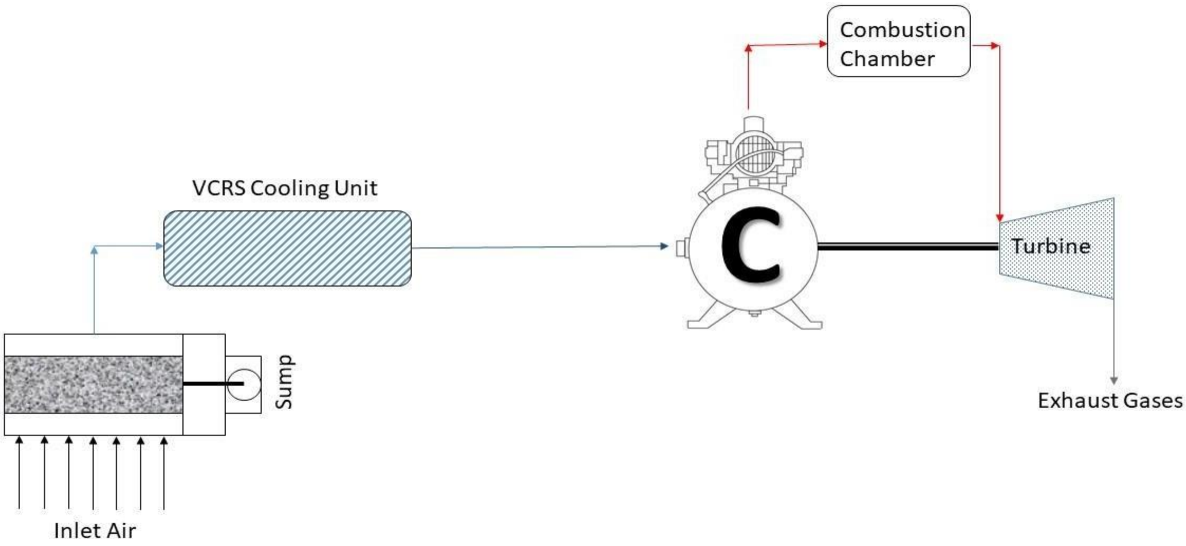

Figure 3.10 Block diagram for the integrated cooling system

As seen in fig 3.10 integrated cooling system is a combination of two cooling systems (Evaporative cooling system and Mechanical cooling system). The main purpose of integrating two different cooling systems is to use the benefits to both systems without there drawbacks.

Evaporative cooling system is a very low capital cost cooling system along with low operating and maintenance cost and simple in design & installation but minimum temperature achieve after cooling is constrained by ambient air condition which is wet bulb temperature corresponding to ambient condition, so to eliminate this drawback Mechanical cooling system is used in which minimum achieve temperature is not constrained by ambient air condition so any desired temperature is obtained, but the capital cost of the mechanical cooling system is high so to achieve the desired temperature with a low capital cost both system used as the integrated cooling system.

An integrated cooling system air is entered in the evaporative cooling system from an ambient condition where heat is the transfer from hot ambient air to flowing water and ambient air cool down wet bulb temperature if the evaporative cooling system has 100% effectiveness and near to wet bulb temperature if the evaporative cooling system does not have 100% effectiveness and if desired CIT is below wet bulb temperature than air is passed through the mechanical cooling system after evaporative cooling system, in mechanical cooling system cooling is done with VCRS unit in which a refrigerant flow in a cycle between compressor, condenser, expansion valve, & evaporator and cool air to desired CIT. After that air enters into the gas turbine's compressor where it is compressed to high pressure and temperature and allows to entry in the combustion chamber where heat is added after burning of fuel and the amount of fuel to be burn depends on the desired TIT. This burn fuel and air mixture are allowed to expand in the turbine where work is produced to rotate the generator and produce electricity.

Thermodynamics analysis of the integrated cooling system is done with the help of the equation used in the evaporative cooling system and mechanical cooling system in 3.3 and 3.4 with all assumptions and input parameters described in Table 2 &3 and obtains results are explained in chapter 4, result and discussion.

CHAPTER 4

RESULTS AND DISCUSSION

Based on the thermodynamics analysis the following results have been obtained and explain here for studying the effect of turbine inlet temperature, compressor inlet temperature, compression ratio, the ambient temperature on gas turbine performance.

VARIATION OF DIFFERENT PARAMETER WITH AMBIENT TEMPERATURE.

Variation of specific net-work output with ambient temperature.

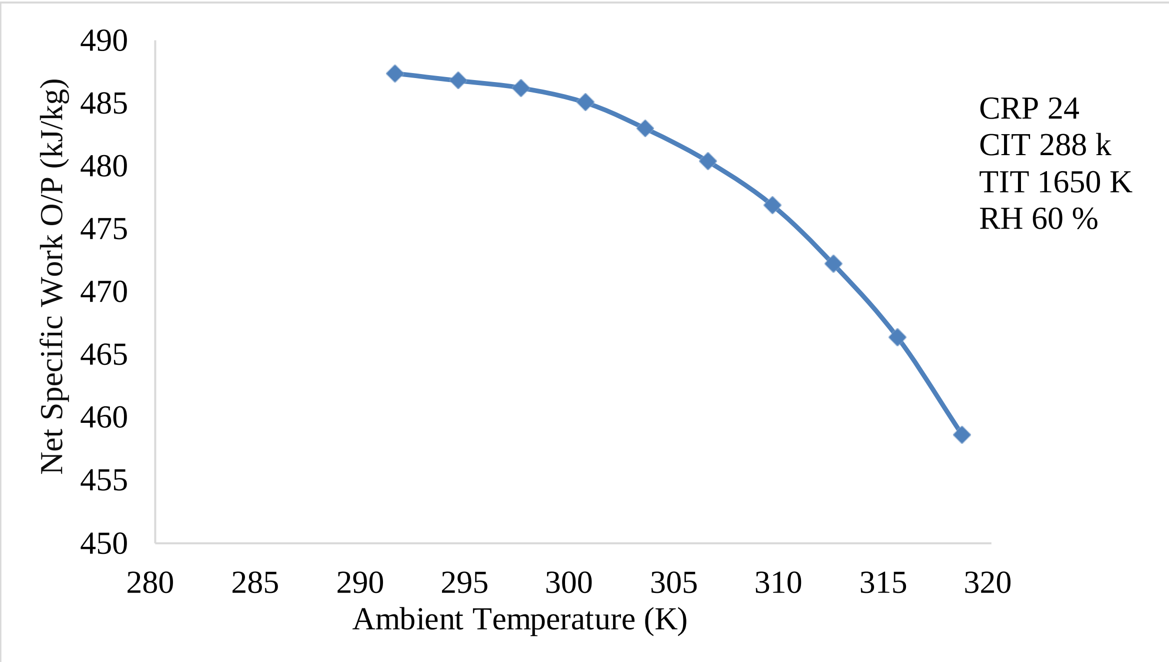

Figure 4.1 Variation of specific net-work output with ambient temperature.

Fig 4.1 tells the variation of specific net-work output with ambient temperature and from fig, it can be concluded that net specific work output decreases with an increase in ambient temperature because net specific work output is the algebraic sum of turbine work, compressor work, and compressor work for VCRS unit so when the ambient temperature is increased then the cooling load on VCRS unit increases which further increases the compressor work for VCRS unit which finally leads to decreasing in net specific work output when CPR is 24, CIT is 288K, TIT is 1650K and RH is 60% remains constant. . There is a drop of 28.1 kJ/kg in net specific work done when temperature increase from 293K to 320K.

Variation of total cooling load with ambient temperature.

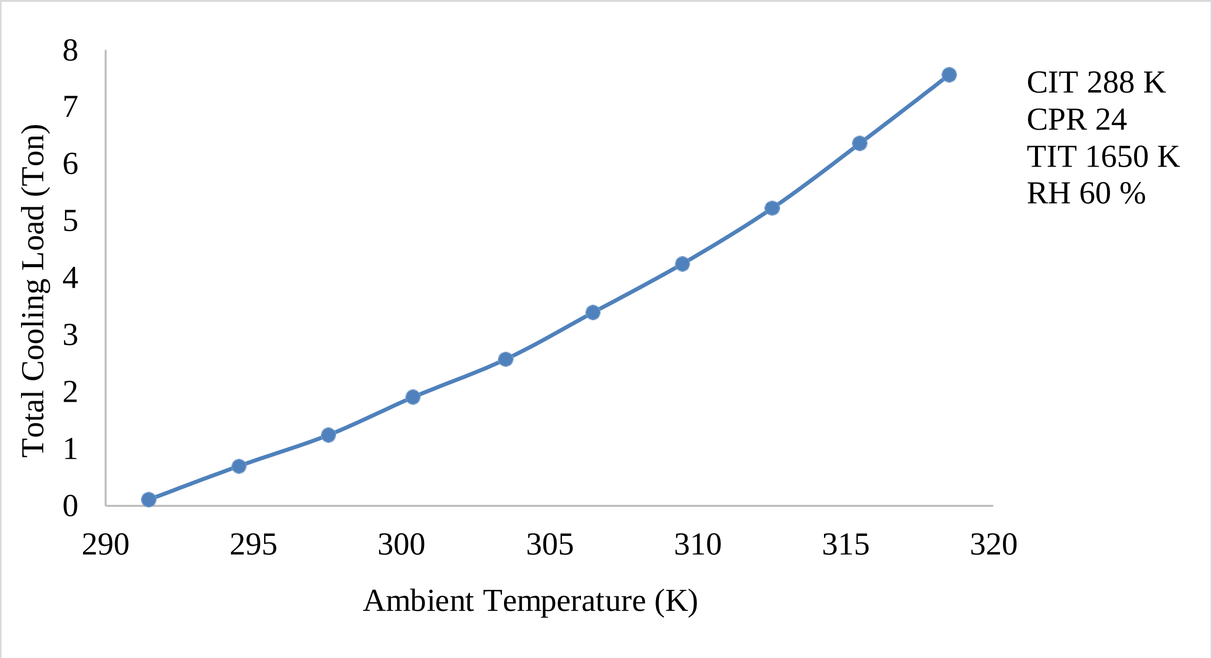

Figure 4.2 Variation of total cooling load with ambient temperature.

From Fig 4.2 it can depict that the total cooling load increases with increasing ambient temperature when CPR at 24, CIT at 288K, TIT at 1650K, and RH at 60% remains constant, this increase in total cooling load can easily be explained with the help of VCRS unit because outlet of VCRS unit and inlet of gas turbine compressor is same which is fixed at 288K so when there is an increase in ambient temperature then there is an increase in inlet temperature of VCRS unit which finally increases the cooling load of VCRS unit. The cooing load increase from 1.6 kJ/s to 27.4 kJ/s when the temperature increase from 293K to 320K.

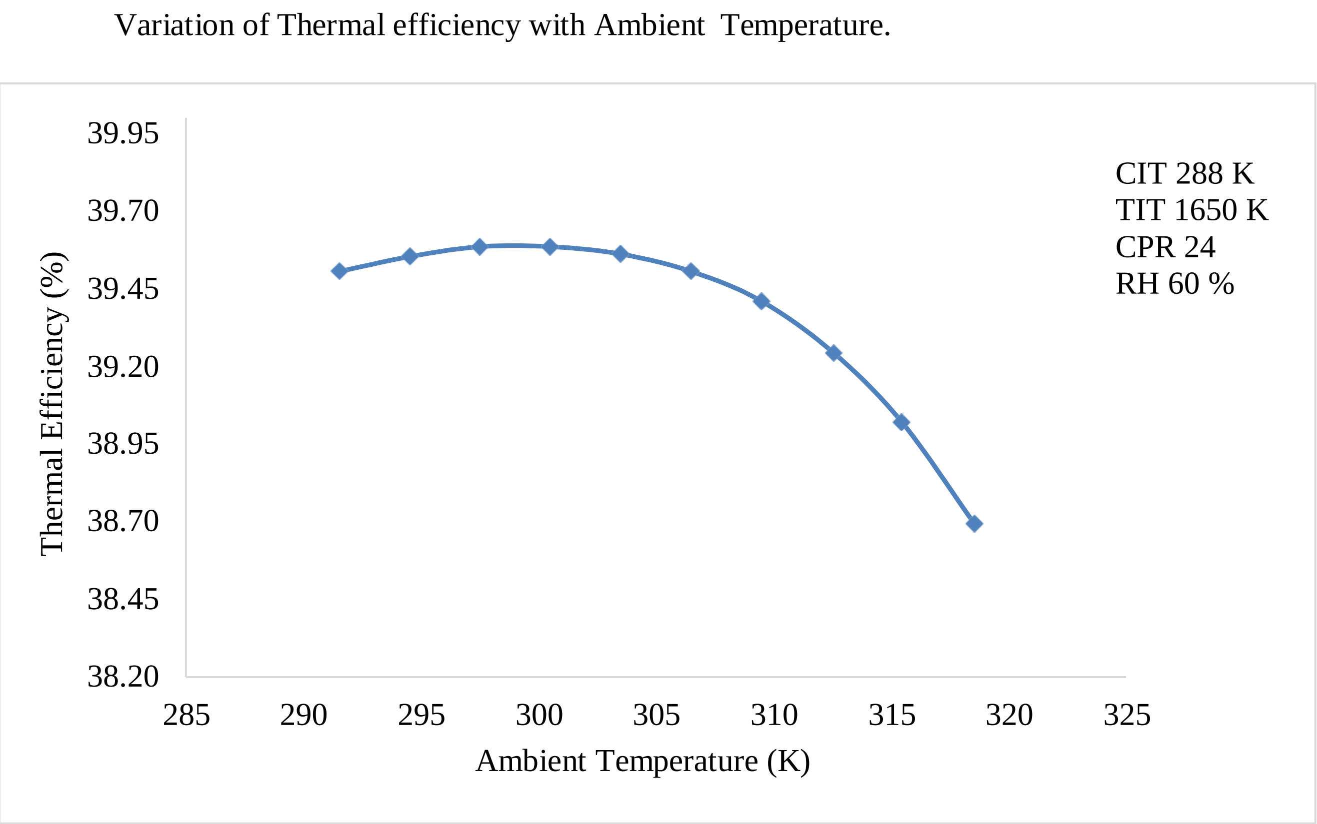

Figure 4.3 Variation of Thermal efficiency with Ambient Temperature.

Fig 4.3 tells the variation of Thermal efficiency with Ambient temperature and it is clear that thermal efficiency decreases when ambient temperature increases, because thermal efficiency is ratio of net specific work output and heat supplied in combustion chamber, for constant turbine inlet temperature of 1650K a constant amount of fuel is supplied in combustion chamber which means thermal efficiency is directly proportional to net specific work output and net specific work output, decreases with increase in ambient temperature so thermal efficiency also decreases with increase in ambient temperature. . When ambient temperature increase from 293K to 320K, thermal efficiency decrease by 2.4%.

Variation of cooling load with Ambient Temperature.

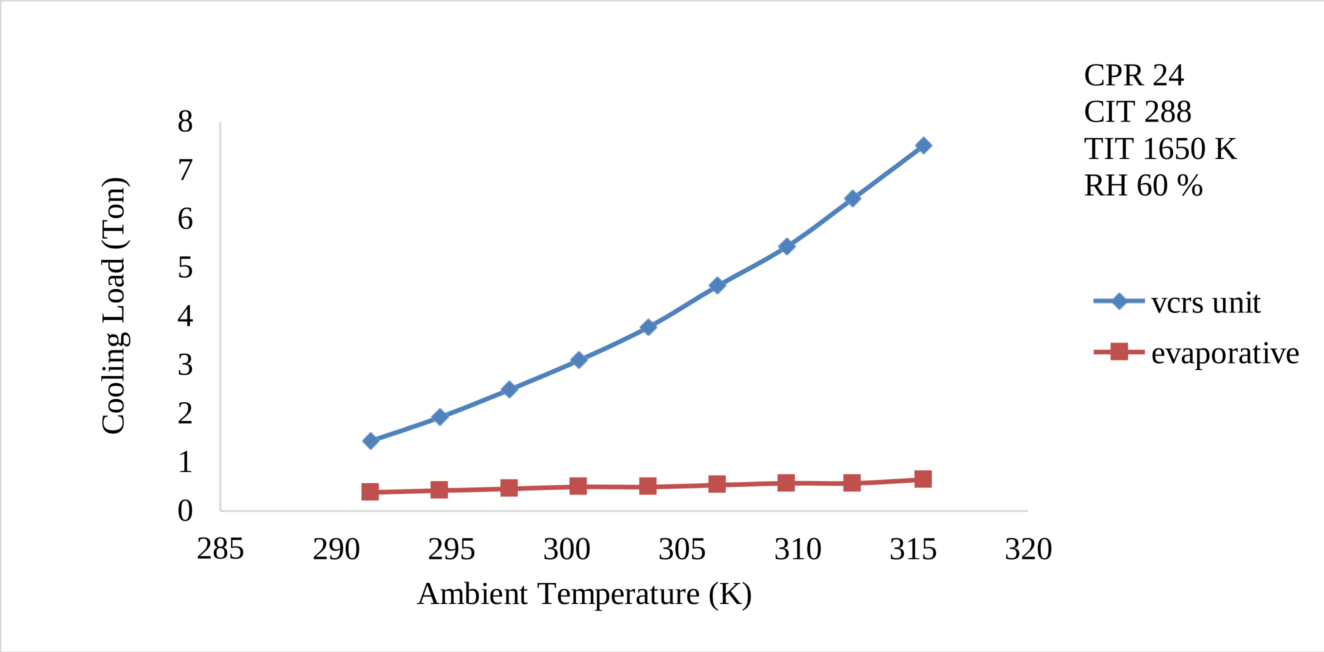

Figure 4.4 Variation of cooling load Ambient Temperature.

Fig 4.4 depicts the cooling load share by two different inlet cooling systems when the ambient temperature is increased. From the graph, it is clear that the total cooling load increases with an increase in ambient temperature, at a lower temperature of 293K evaporative cooling system have 4.145 kJ/s of cooling load, and VCRS unit has 16.89 kJ/s of cooling load which is 75% of total cooling load and at a higher temperature of 317K evaporative cooling system have 7.14 kJ/s of cooling load and VCRS unit have 87.73Kj/s of cooling load which is 87% of total cooling load i.e. with an increase in ambient temperature load share by VCRS unit also increases because in evaporative cooling system minimum temperature attain is wet bulb temperature corresponding to ambient condition.

VARIATION OF DIFFERENT PARAMETER WITH TURBINE INLET TEMPERATURE (TIT).

Variation of net specific work output with Turbine Inlet Temperature.

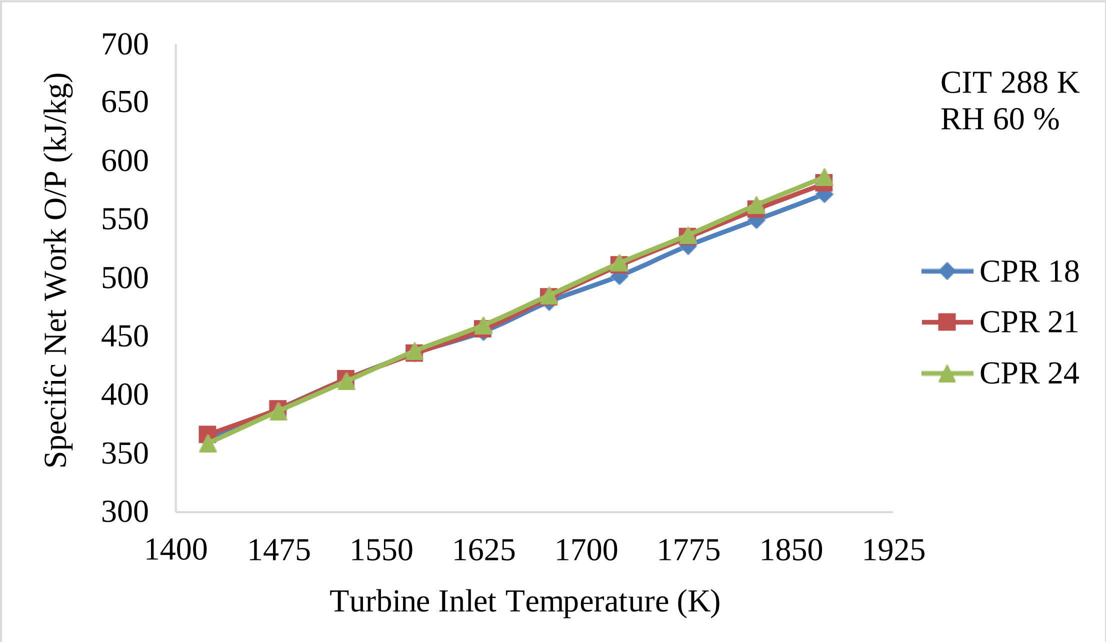

Figure 4.5 Variation of specific net-work output with Turbine Inlet Temperature.

Fig 4.5 depicts the variation of specific net-work output with turbine inlet temperature for different cycle pressure ratio when compressor inlet temperature and atmospheric temperature are at 288K and 318K respectively with ambient relative humidity 60%. From the given graph it is clear that net-specific work output increase with an increase in turbine inlet temperature. This happens because when turbine inlet temperature increase then works done by turbine also increase which further increase the net specific work output of cycle for particular pressure ratio and net specific work output also increase with the increase in pressure ratio so forgiven graph maximum net specific work output is obtained at 1900K turbine inlet temperature and 24 cycle pressure ratio.

At CPR 21, when TIT increases from 1450K to 1900K net specific work output increases by 57%.

Variation of specific fuel consumption (sfc) with Turbine Inlet Temperature.

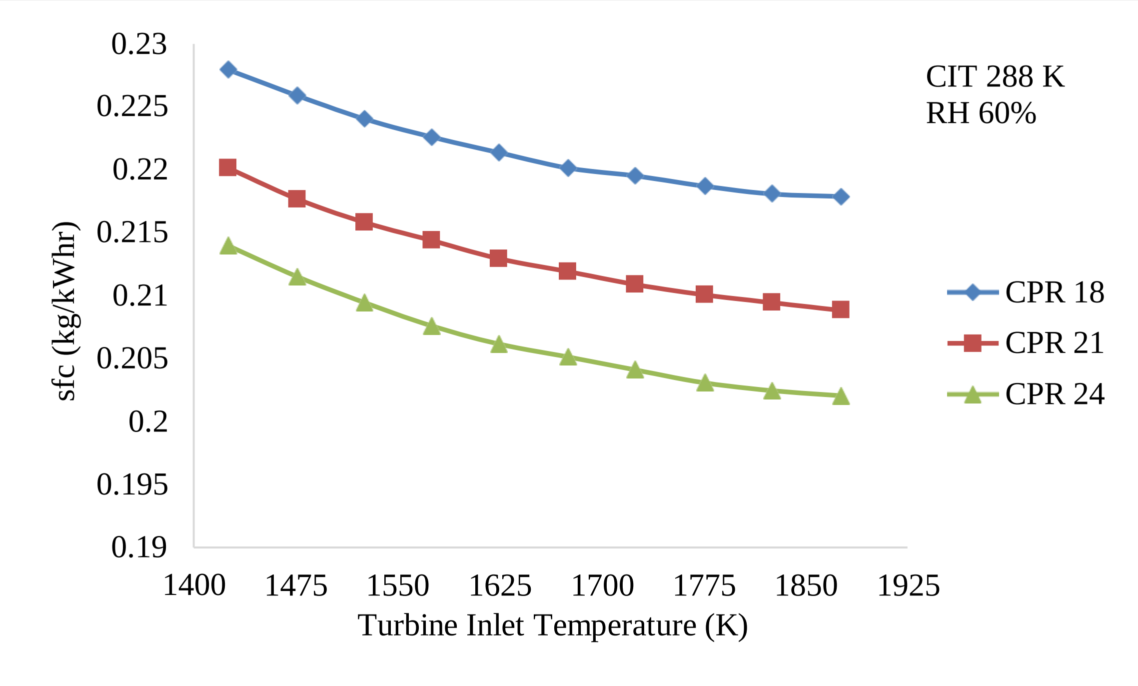

Figure 4.6 Variation of sfc with Turbine Inlet Temperature.

Fig 4.6 clear shows the variation of specific fuel consumption with turbine inlet temperature for different pressure ratios. From the graph it is easily observed that specific fuel consumption decreases with an increase in turbine inlet temperature and this variation is easily explained by specific fuel consumption formula which is the ratio of the mass flow rate of fuel and net specific work output so when there is an increase in turbine inlet temperature then the mass flow rate of fuel also increases because we need more fuel in the combustion chamber to get the higher temperature at combustion chamber exit and net specific work output also increase with the increase in turbine inlet temperature but the rate of increasing net specific work output is more compare to the mass flow rate of fuel so the net effect of the increase in net specific work output and mass flow rate of fuel decreases in specific fuel consumption.

There is a drop in sfc by 4.3% when TIT increases from 1400K to 1900K.

Variation of Thermal Efficiency with Turbine Inlet Temperature.

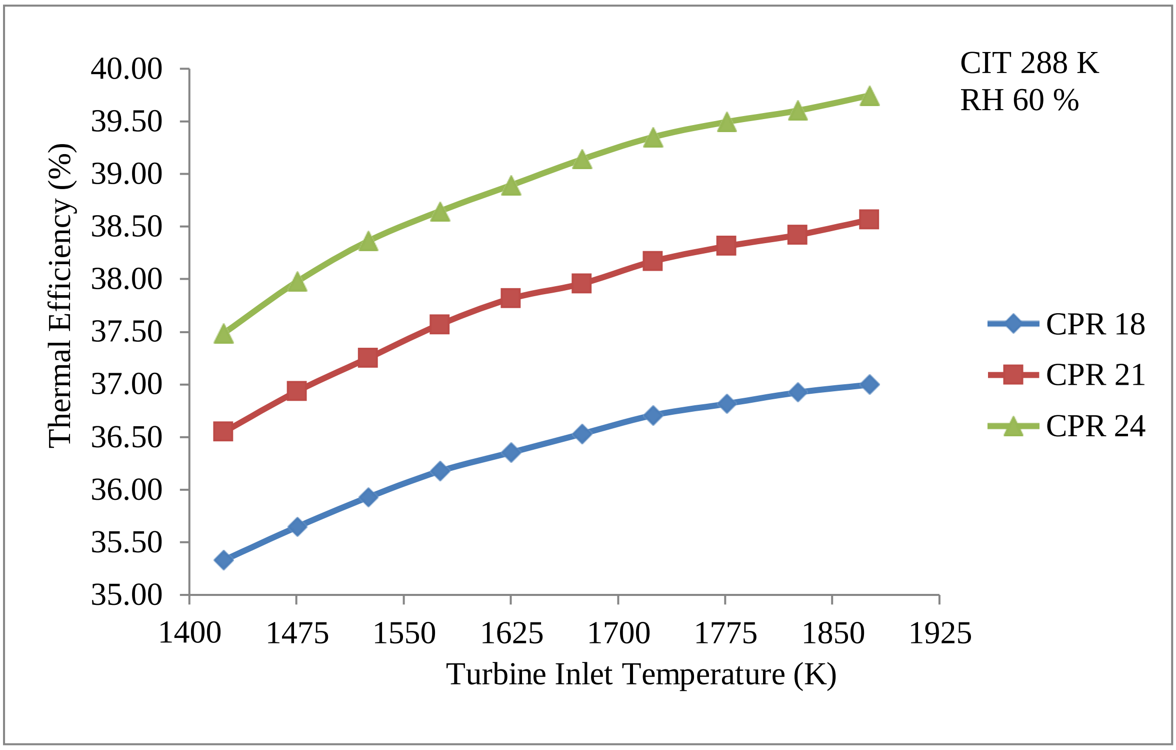

Figure 4.7 Variation of Thermal Efficiency with Turbine Inlet Temperature.

Fig 4.7 depicts the variation of thermal efficiency with turbine inlet temperature at a compressor inlet temperature of 288K and relative humidity of 60% for different pressure ratios. The thermal efficiency of the cycle increase with an increase in turbine inlet temperature and also increases with an increase in pressure ratio. Thermal efficiency is the ratio of net specific work output and heat added in the combustion chamber so when turbine inlet temperature increases then turbine work output increases which further increases net specific work output and heat addition in the combustion chamber also increases to get the high temperature at turbine inlet or at combustion chamber exit. But the rate of increasing net-specific work output is more compare to the rate of heat addition in the combustion chamber so the overall effect of increasing turbine inlet air temperature is an increase in the thermal efficiency of the plant.

Thermal efficiency increase by 4.3% when TIT increase from 1450K to 1900

VARIATION OF DIFFERENT PARAMETER WITH COMPRESSOR INLET TEMPERATURE (CIT).

Variation of compressor work with compressor inlet temperature.

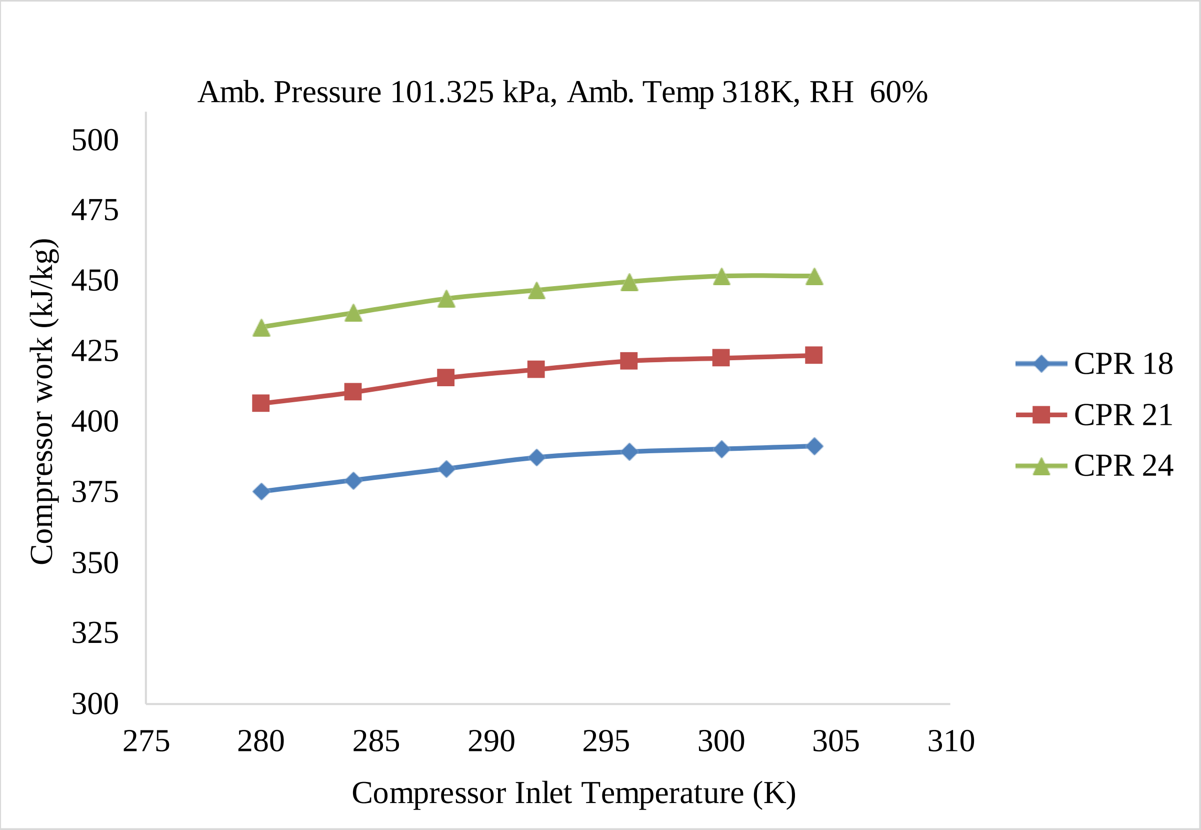

Figure 4.8 Variation of compressor work with compressor inlet temperature.

Fig 4.8 depicts the variation of compressor work for different compressor inlet temperature when ambient pressure and ambient temperature are fixed at 101.325kPa and 318K respectively. From the graph, it is clearly seen that compressor work is increasing with an increase in compressor inlet temperature because compressor work is directly proportional to compressor inlet temperature and decrease and relative humidity with an increase in compressor inlet temperature is another reason for the decrease in compressor work. Compressor work also increases with an increase in the compression pressure ratio.

For CPR 21 when the compressor inlet temperature increase from 282K to 306K compressor work increase by 3.9%.

Variation of specific net-work output with compressor inlet temperature.

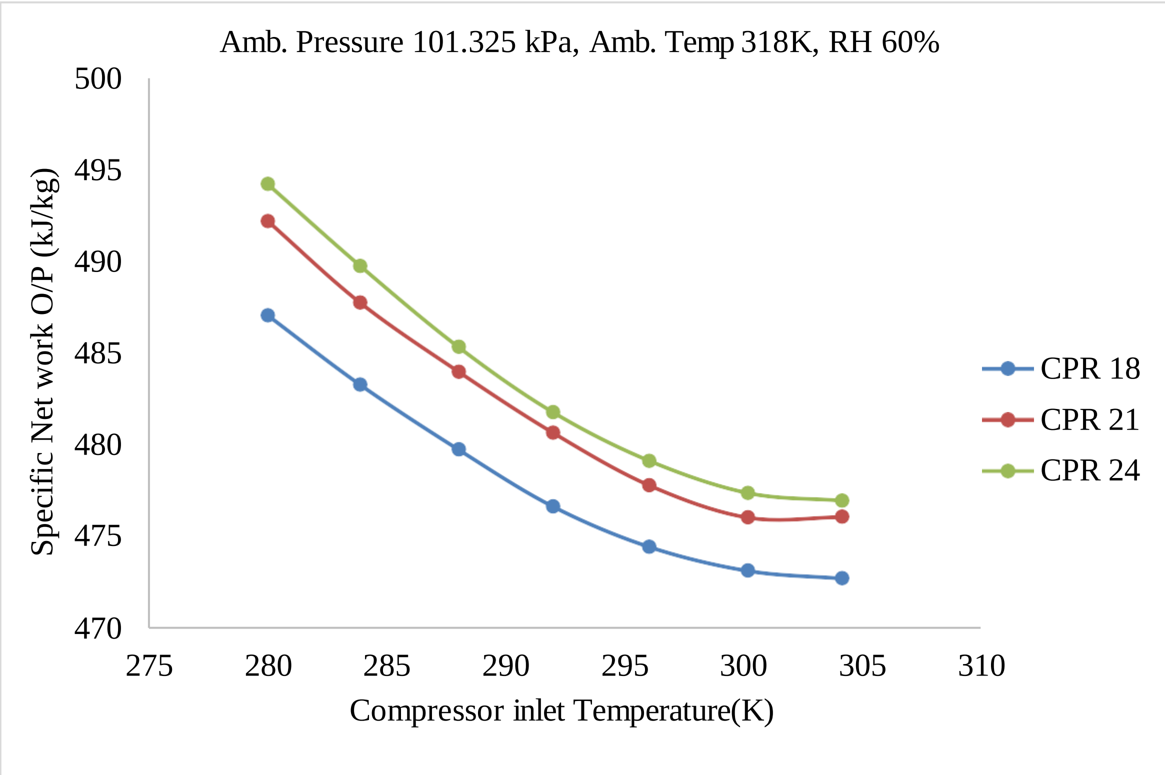

Figure 4.9 Variation of specific net-work output with compressor inlet temperature.

From Fig 4.9 it can be concluded that net-specific work output decrease with an increase in compressor inlet temperature. Net specific work output is the algebraic sum of turbine work done, compressor work done, and compressor work done of VCRS unit. For turbine inlet temperature of 1650K, turbine work done is constant and the variation of compressor work done is explain in fig 24 but the compressor work done of VCRS unit also increases with compressor inlet temperature which decreases net specific work output and this variation can be computed for different pressure ratio.

For CPR 21, there is a 2.5% drop in net specific work when the compressor inlet increase from 282K to 306K.

Variation of Thermal Efficiency with compressor inlet temperature.

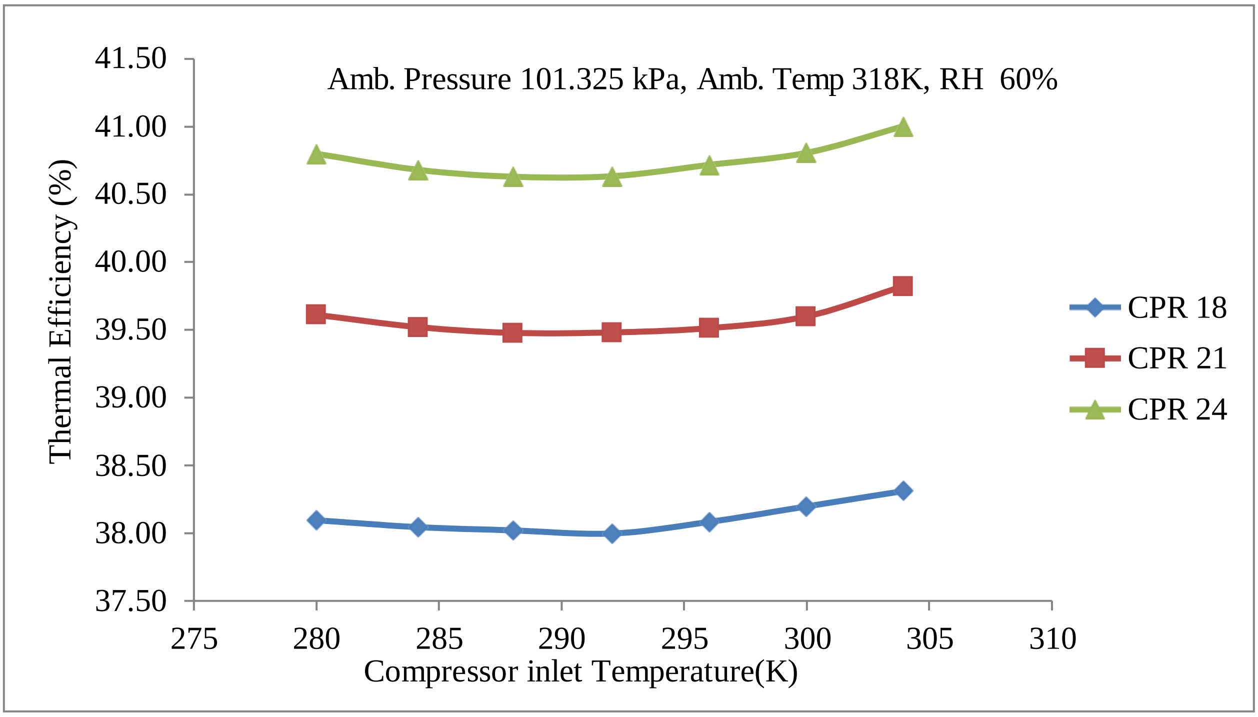

Figure 4.10 Variation of Thermal Efficiency with compressor inlet temperature.

Fig 4.10 depicts the variation of thermal efficiency with compressor inlet temperature and shows that thermal efficiency decreases with an increase in compressor inlet temperature. Thermal efficiency is the ratio of net specific work output and heat added in the combustion chamber, for constant turbine inlet temperature heat added in the combustion chamber is constant which means thermal efficiency is directly proportional to net specific work output so the variation of thermal efficiency vs compressor inlet temperature is same as variation of net specific work output vs compressor inlet temperature.

For CPR 21, when compressor inlet temperature increase from 282K to 306K thermal efficiency 0.5%.

VARIATION OF DIFFERENT PARAMETER WITH AMBIENT RELATIVE HUMIDITY

Variation of Net Specific Work Output with Relative Humidity.

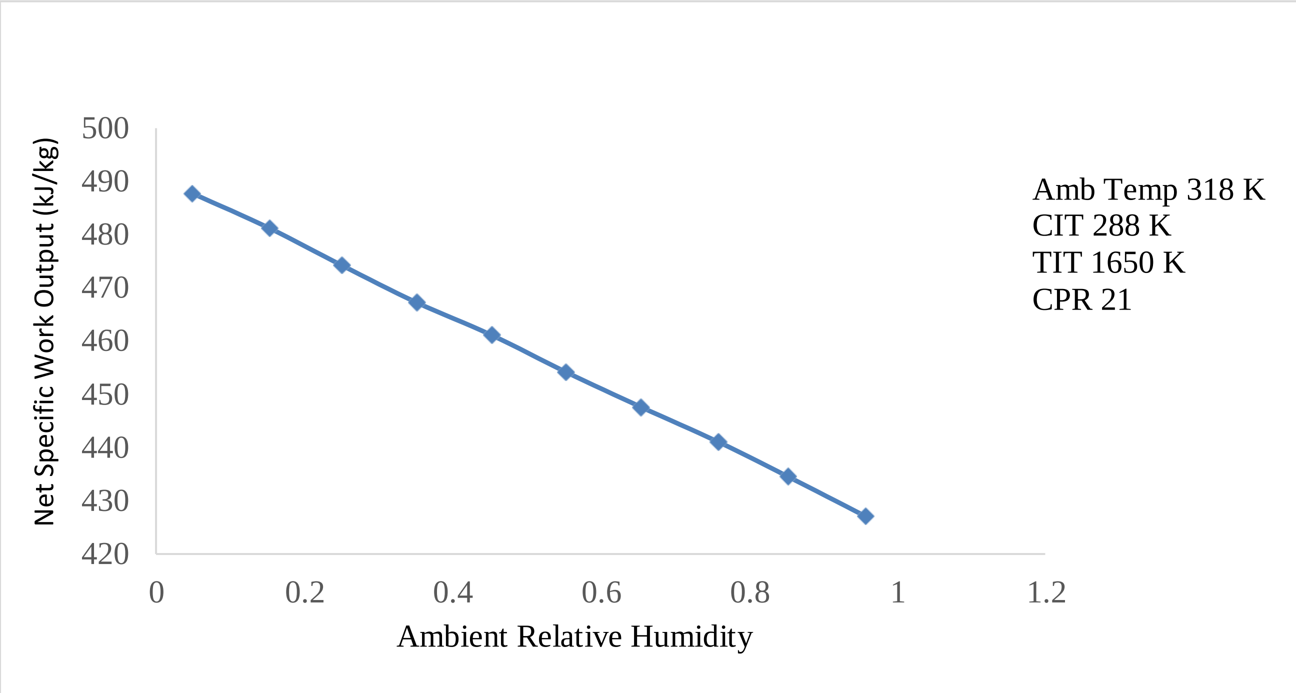

Figure 4.11 Variation of net specific work output with ambient relative humidity.

Fig 4.11 depicts the variation of specific net-work output with ambient relative humidity and tells that net specific work output decreases with an increase in ambient relative humidity this happens because when ambient relative humidity increases then wet bulb temperature also increases which means the exit temperature of evaporative cooler increases so for a particular compressor inlet temperature, VCRS unit has to done more cooling which means more work is to be done by VCRS unit compressor which decreases the net specific work output.

When ambient relative humidity increase from 0.12 to 1 there is a drop of 57.9 kJ/kg in net specific work done.

Variation of Thermal Efficiency with Ambient Relative Humidity

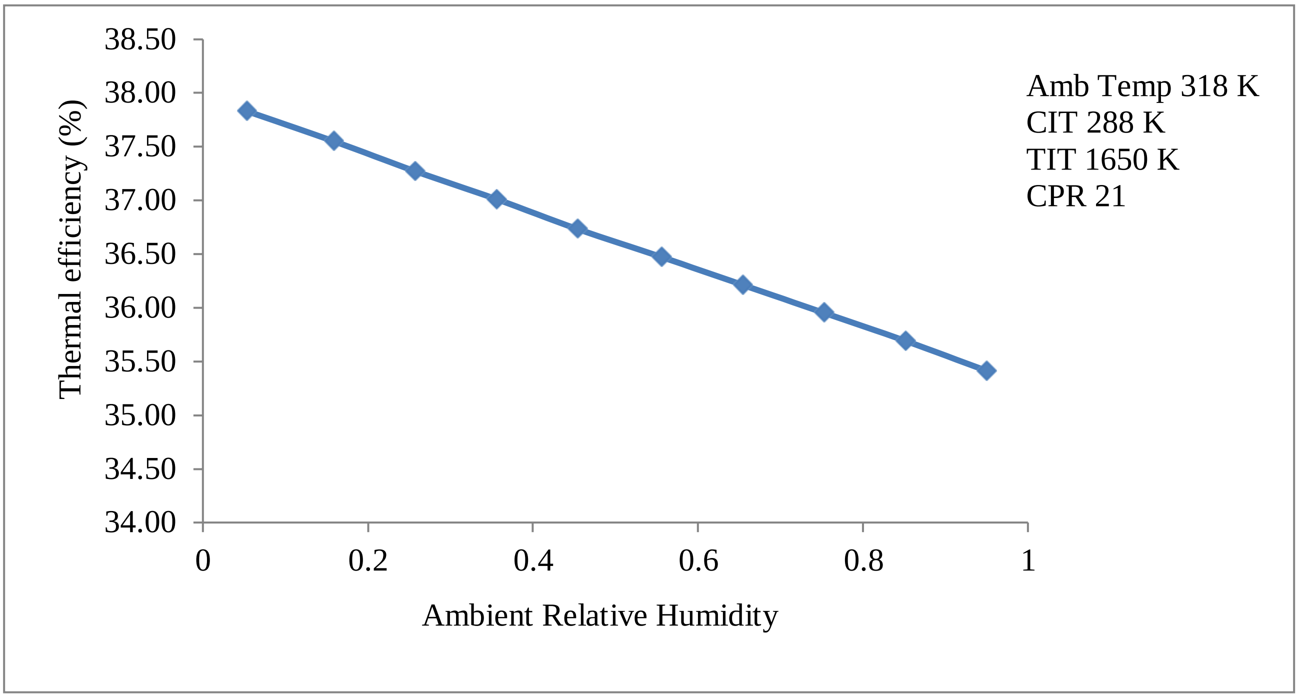

Figure 4.12 Variation of Thermal efficiency with Ambient Relative Humidity

Fig 4.12 depicts the variation of thermal efficiency with ambient relative humidity and tells that thermal efficiency decreases with an increase in ambient relative humidity this happens because when ambient relative humidity increases then wet bulb temperature also increases which means the exit temperature of evaporative cooler increases so for a particular compressor inlet temperature, VCRS unit has to done more cooling which means more work is to be done by VCRS unit compressor which decreases the net specific work output. And thermal efficiency is the ratio of net specific work output and heat added in the combustion chamber, for particular TIT heat added in the combustion chamber is constant so for this case thermal efficiency is directly proportional to net specific work done so with a decrease in net specific work output thermal efficiency also decrease which can be seen in fig x.

When ambient relative humidity is an increase from 0.12 to 1, thermal efficiency decrease from 0.35 to 0.33.

VARIATION OF DIFFERENT PARAMETER WITH AMBIENT TEMPERATURE FOR VARIOUS GAS TURBINE COOLING SYSTEM.

In this section variation of the different parameters with ambient temperature is discussed for various gas turbine cooling systems. Comparison is made between Simple Gas Turbine(SGT), Simple Gas Turbine with Inlet Fogging(SGT+IF), Simple Gas Turbine with Mechanical Cooling(SGT+MC), and an integrated system of Simple Gas Turbine with Inlet Fogging and Mechanical Cooling(SMC+IF+MC).

Comparison is made by taking the same input parameter for all system and cooling systems are achieving the minimum possible desired temperature which is wet bulb temperature corresponding to ambient air for the evaporative system and 288K for the mechanical cooling system.

Variation of Net Specific Work Output with Ambient Temperature for various gas turbine cooling system

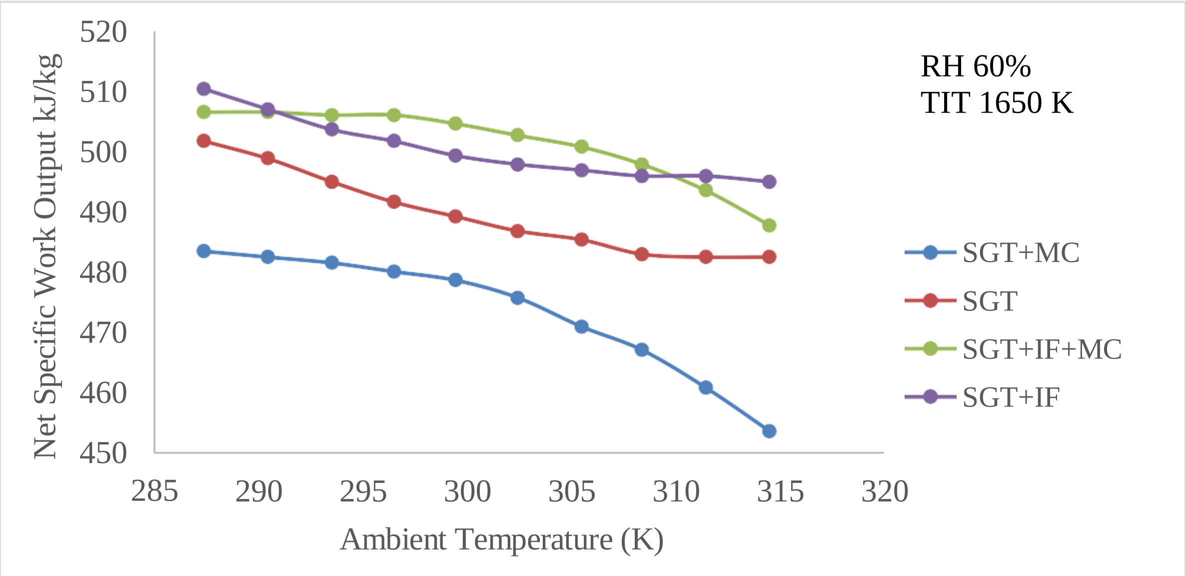

Figure 4.13 Variation of Net Specific Work Output with Ambient Temperature for gas various turbine cooling system.

From the fig 4.13, it can be seen that net specific work output is decreased with an increase in ambient temperature for all system but the integrated system (SGT+IF+MC) has maximum net specific work output among other systems at 300K that means gas turbine with integrated cooling system give maximum net-work output at 300K.

Variation of Cooling Load with Ambient Temperature for various gas turbine cooling system.

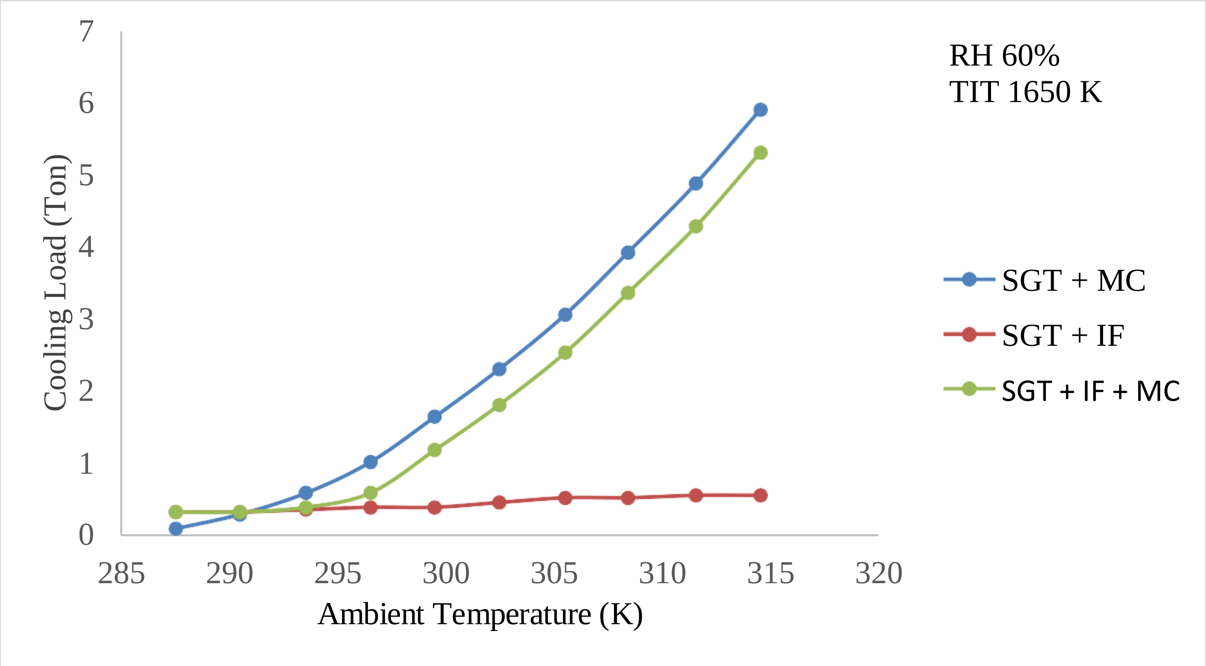

Figure 4.14 Variation of Cooling load with Ambient Temperature for various gas turbine cooling system

From fig 4.14 it can be depicted that cooling load is almost constant at lower temperature up to 300K and start increasing drastically for the mechanical cooling system and integrated system after 300K and remains almost constant for an evaporative cooling system. For the evaporative cooling system minimum, the attainable temperature is wet bulb temperature corresponding to ambient condition so the evaporative cooling system has less cooling load, on the other hand, the mechanical cooling system has more cooling load compared to the integrated cooling system which means mechanical cooling system compressor done more work which increases the operating cost of the mechanical cooling system in comparison to the integrated cooling system which implies that integrated cooling is more cost-effective.

There is an average difference of 7.5kJ/kg in the cooling load of the mechanical system and integrated system after 300K.

Variation of Thermal Efficiency with Ambient Temperature for various gas turbine cooling system.

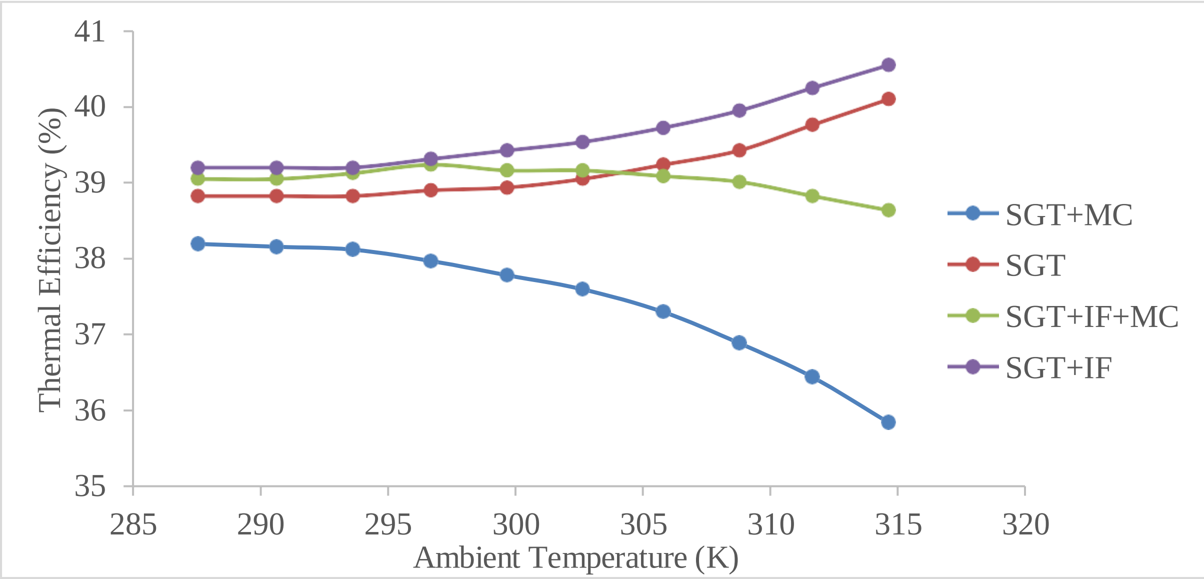

Figure 4.15 Variation of Thermal Efficiency with Ambient Temperature for various turbine cooling system.

From fig 4.15 it can be easily observe that thermal efficiency of SGT and SGT with IF increase with temperature because with increase in ambient temperature compressor inlet temperature increases which further increase the compressor outlet temperature which decrease the heat addition in combustion chamber and mass flow rate of fuel which finally decrease the efficiency for SGT and in SGT with IF compressor inlet temperature decreases because of IF which decrease the compressor work done and increase the net specific work output which finally increase the thermal efficiency and in SGT with MC efficiency decrease with ambient temperature because with increase in ambient temperature cooling load increases which increase compressor work done of VCRS unit and decrease the net specific work done and efficiency but in SGT with IF and MC efficiency is increase with ambient temperature and varies between the range of efficiency of SGT and SGT with IF because cooling load is share by both cooling system and reduce the cooling cost for integrated system.

CHAPTER 5

CONCLUSION

Based on the thermodynamics analysis on SGT integrated with the various cooling system following conclusions have been drawn.

- An increase in ambient temperature degrades the net specific work and thermal efficiency of the gas turbine and increases the cooling load. There is a 4.7% decrease in net-specific work output, a 51% increase in cooling load, and a 22% decrease in thermal efficiency when ambient temperature increase from 293K to 320K.
- Gas turbine performance improves with an increase in TIT. Net specific work output increase by 59%, thermal efficiency increase by 4.8%, and SFC decrease by 5.2% when TIT increase from 1450K to 1900K at CPR 21.
- Compressor work increases with an increase in CIT by 4.2% at CPR of 21 and efficiency increases by 0.5 % with a decrease in net specific work output by 3.4%.
- When relative humidity increase from 1% to 100% net specific work decrease by 14% and thermal efficiency decrease by 7%.
- By comparing the various cooling system with the integrated cooling system it can be concluded that the integrated cooling system gives maximum net specific work O/P with minimum cooling load and moderate efficiency.
- From all the results it can be concluded that an integrated cooling system is one of the best cooling systems for the hot and dry region.

REFERENCES

[1] Annual Report 2016-17 & 2017-18,Ministry of Power , Government of India.

[2] Al-Ibrahim, A.M. and Varnham, A., 2010. A review of inlet air-cooling technologies for enhancing the performance of combustion turbines in Saudi Arabia. *Applied thermal engineering*, *30*(14-15), pp.1879-1888.

[3] MacCracken, C.D., 1994. Overview of the progress and the potential of thermal storage in off-peak turbine inlet cooling. *ASHRAE transactions*, *100*(1), pp.569-571.

[4] Kitchen, B.J. and Ebeling, J.A., 1995, June. Qualifying combustion turbines for inlet air cooling capacity enhancement. In *Turbo Expo: Power for Land, Sea, and Air* (Vol. 78811, p. V004T10A007). American Society of Mechanical Engineers.

[5] De Lucia, M., Lanfranchi, C. and Boggio, V., 1996. Benefits of compressor inlet air cooling for gas turbine cogeneration plants.

[6] Stewart Jr, W.E., 1999. Designing for combustion turbine inlet air cooling. *ASHRAE Transactions*, *105*, p.165.

[7] Chaker, M., Meher-Homji, C.B. and Mee III, T., 2002, January. Inlet fogging of gas turbine engines: Part A—fog droplet thermodynamics, heat transfer and practical considerations. In *Turbo Expo: Power for Land, Sea, and Air* (Vol. 36096, pp. 413-428).

[8] Kim, K.H., Ko, H.J., Kim, K. and Perez-Blanco, H., 2012. Analysis of water droplet evaporation in a gas turbine inlet fogging process. *Applied Thermal Engineering*, *33*, pp.62-69.

[9] Mohapatra, A.K., 2015. Comparative analysis of inlet air cooling techniques integrated to cooled gas turbine plant. *Journal of the Energy Institute*, *88*(3), pp.344-358.

[10] Carmona, J., 2015. Gas turbine evaporative cooling evaluation for Lagos–Nigeria. *Applied Thermal Engineering*, *89*, pp.262-269.

[11] Barigozzi, G., Perdichizzi, A., Gritti, C. and Guaiatelli, I., 2015. Techno-economic analysis of gas turbine inlet air cooling for combined cycle power plant for different climatic conditions. *Applied Thermal Engineering*, *82*, pp.57-67.

[12] Shukla, A.K. and Singh, O., 2016. Performance evaluation of steam injected gas turbine based power plant with inlet evaporative cooling. *Applied Thermal Engineering*, *102*, pp.454-464.

[13] Rahman, A.A.A. and Mokheimer, E.M., 2017. Boosting gas turbine combined cycles in hot regions using inlet air cooling including solar energy. *Energy Procedia*, *142*, pp.1509-1515.

[14] Shukla, A.K., Sharma, A., Sharma, M. and Mishra, S., 2018. Performance improvement of simple gas turbine cycle with vapor compression inlet air cooling. *Materials Today: Proceedings*, *5*(9), pp.19172-19180.

[15] Mostafa, M., Eldrainy, Y.A. and EL-Kassaby, M.M., 2018. A comprehensive study of simple and recuperative gas turbine cycles with inlet fogging and overspray. *Thermal Science and Engineering Progress*, *8*, pp.318-326.