Thermodynamics Analysis of Integrated Cooling System for Gas Turbine

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CANDIDATE'S DECLARATION

I, Dheeraj Pratap Singh, 2K17/THE/05 student of M.Tech (Thermal Engineering), hereby declare that the project Dissertation title "Thermodynamics Analysis of Integrated Cooling System for Gas Turbine" 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, 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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CERTIFICATE

I hereby certify that the Project Dissertation titled "Thermodynamics Analysis of Integrated Cooling System for Gas Turbine" which is submitted by Dheeraj Pratap Singh, 2K17/THE/05, 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

The Power output of a gas turbine is directly proportional to and limited by the mass flow rate of compressed air available to it from the air compressor that provides high-pressure air to the combustion chamber of the gas turbine system. An air compressor has a fixed capacity for handling a volumetric flow rate of air. Even though the volumetric capacity of a compressor is fixed, the mass flow rate of air it delivers to the gas turbine changes with changes in ambient air temperature. This mass flow rate of air decreases with increase in ambient temperature because the air density decreases when air temperature increases. Therefore, the power output of a combustion turbine decreases below its rated capacity when there is an increase in ambient temperature so this paper deal with different inlet air cooling system incorporated in gas turbine. The objective of this study is to integrate two different cooling systems (evaporative cooling system and mechanical cooling system). Thermodynamics modeling is carried out and presented along with results showing influence of integrated cooling system on various performance parameter of gas turbine and their key benefits and drawbacks in relation to the environmental conditions. The integration of inlet air cooling system reduces the work required to compress the air and increases the net work output of gas turbine and also increases the overall efficiency of gas turbine. The simulation is also carried out by varying different parameters.

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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	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	Coolent
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	Ratio of mass flow rate of fuel to 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

1.1 INTRODUCTION OF COOLING SYSTEM

Energy play a very important role in growth of any country so for any country one of the for most concern is to meet the demand of energy required. With passage of time the demand of energy is increasing with population, in India in 2016-17 electricity generation target for conventional energy sources was 1178 Billion unit which increased to 1229.4 Billion unit in 2017-18[1]. To tackle this increasing demand of energy either number of power plant to be increase or turbine inlet air cooling system can be used which can reduce the burden of increasing energy demand to certain extent and cost of installing inlet air cooling system is low compare to installing a new power plant unit.

Net work output of gas turbine is strongly depends on ambient air temperature i.e. entry point of compressor because the work consume by compressor is directly depend on the temperature of ambient air which is to be compress so when ambient air temperature is more then, compressor consume more power and net power decreases. Cooling the inlet air increase the net power output and improve the efficiency. These inlet air chilling system are more effective in hot and dry region where ambient temperature is high.

Gas turbines are frequently used to meet peak demand during on-peak power periods. Unfortunately, this period usually coincides along the hottest temperatures of the day, that highly affect the gas turbine output, this loss in turbine output due to more ambient air temperature can be compensate by using inlet air cooling.

Inlet air cooling is cooling of the ambient air before it enters the compressor of gas turbine that supplies high-pressure & high temperature air to the combustion chamber from which hot air at high pressure enters the combustion turbine. Inlet air cooling also called by many others name, including combustion turbine inlet air cooling, turbine inlet air cooling, combustion turbine air cooling and gas turbine inlet air cooling.

Inlet air cooling systems are not only the best solution for a peak demand period (capital cost is around one third the capital investment of a new gas turbine), but also improve the gas turbine heat rate. By improving the heat rate, inlet air cooling system increases the power per unit of fuel, thus making the gas turbine more efficient and reducing carbon emissions.

1.2 PRINCIPLE

The Power output of a gas turbine is directly proportional to the mass flow rate of compressed air getting from the air compressor that supply high pressure air for the combustion chamber of the gas turbine system. All air compressor have a fixed capacity for handling a volumetric flow rate of air at a particular RPM. Even though the volumetric handling capacity of all compressor is fixed, the mass flow rate of air it delivers to the gas turbine changes with changes in atmospheric air temperature. This mass flow rate of air drops with increase in ambient temperature because the air density decreases when air temperature increases. Therefore, the power output of a gas turbine drops below its rated capacity at the ISO conditions (15°C, relative humidity 60%, and absolute pressure(sea level) 101.325kPa) with increases in ambient temperature above 15°C. inlet air cooling system allows increase in air density by decreasing the temperature and thus, helps increase mass flow rate of air into the gas turbine and results in improve output of the gas turbine.

1.3 BENEFITS OF INLET AIR COOLING SYSTEM

- Net Power output of gas turbine increase.
- Also used to cool auxiliary component.
- Turbine heat rate improve.
- Constant inlet air temperature.
- Avoid gas turbine output fluctuation produce by ambient condition.
- Lower price per generated MW compare to new gas turbine plant.
- Reconognized green technology with carbon credits.

1.4 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 energys storage

1.4.1 WETTED MEDIA EVAPORATIVE COOLING

As the name suggest in wetted media evaporative cooling, cooling is done with the help of evaporation. This system comprises of air filter, wetted media, water storage tank, water pump. First ambient air is allow to pass through air filter where all unnecessary debris are separated after that filter air is allow to move through wetted media in which water is coming from one side and filter air is coming from other side, wetted media is a wet porous surface, there are many type of wetted media system and honey comb is best among them. This type of system is good for hot and dry regions.

In wetted media when wet surface and hot air come in contact then some water from wet surface take heat from hot air and evaporate and decrease the temperature of inlet hot air, in this process sensible heat loss from inlet air act as latent heat gain for water and this whole process is isenthalpic process along with cooling and humidification process as psychrometric process.

Because of continuous evaporation of water from wetted media, the salt concentration in water increases which increases the chances of scaling so a purification system is required and a continuous supply of water is also

required to compensate the evaporated water, mass of water requires is depend on degree of coolings required, ambient condition and turbine mass flow rate.
The major drawback of this system is that this system can not decreases temperature of hot inlet air below wet bulb temperature corresponding to ambient condition.

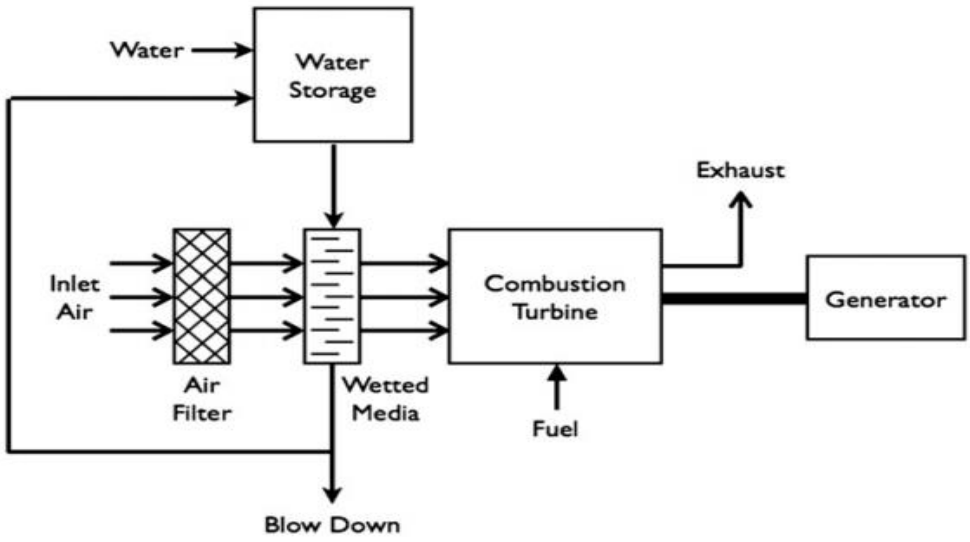

Figure 1.1 Turbine inlet air cooling system using wetted evaporative system [2]

1.4.2 FOGGING

In this system principal of cooling is evaporation of water and water is provided in atomized form with the help of nozzles. This system comprises of demineralized water plant, demineralized water storage tank, inlet air filter, fogging spray (array of nozzles) and gas turbine.
First water is demineralized to reduce the chance of fouling at compressor blade, usually mist eliminator is installed after fogging spray to remove the unevaporated water droplets which can damage the compressor blade.
In fogging when fine water droplets and hot air come in contact then water droplet take heat from hot air and evaporate and decrease the temperature of

inlet hot air, in this process sensible heat loss from inlet air act as latent heat gain for water and this whole process is isenthalpic process along with cooling and humidification process as psychrometric process.

Designing of fogging system done by keeping in mind that droplets size should be near the 20 micron to increase the evaporation rate and duct length between fogging spray and compressor is such that the time taken by water droplet to cover this distance should be more than time required to evaporate and evaporation time is depends on diameter of droplet, DBT, WBT.

This system have some disadvantages like its highly dependency on relative humidity, it required demineraziled water, and minimum temperature obtain is wet bulb temperature corresponding to ambient condition.

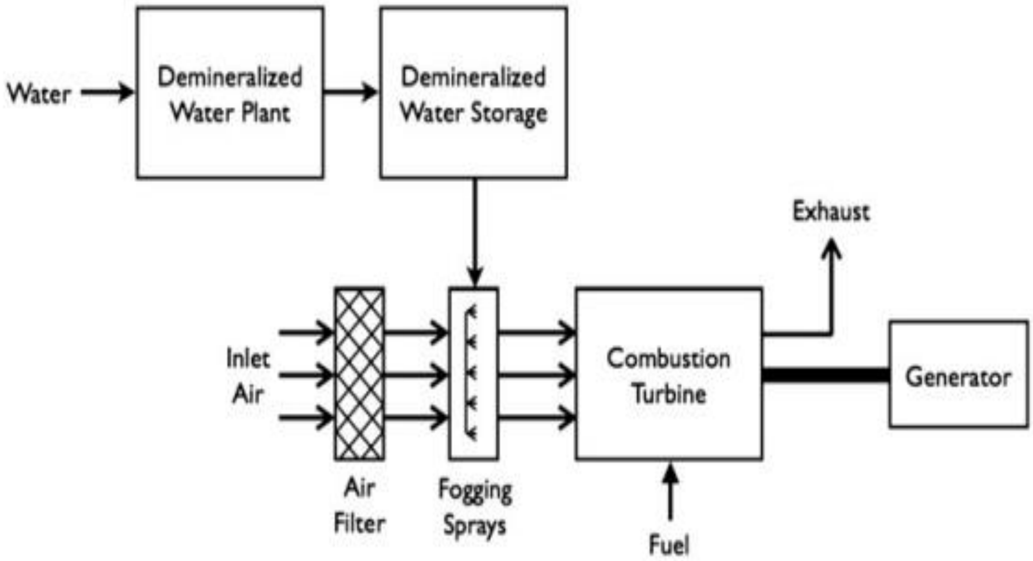

Figure 1.2 Turbine inlet air cooling using fogging. [2]

1.4.3 WET COMPRESSION / OVER SPRAY

In wet compression water droplet is added to the inlet air as a fog just as it done for fogging. However, the amount of fog added is a lot more than that can be evaporated under the conditions of the ambient air. The inlet air stream carries the extra water into the compressor sections of the gas turbines from where it evaporates further, cools the compressed air and develop more mass for boosting the gas turbines output beyond the possible with the help of evaporative cooling technologies. The amount of excess mist carried into the compressor depends on where the fog is added in the inlet section of the gas turbine system. This is called continuous cooling because cooling continuous during compression also because of evaporation of extra fog.

Compare to fogging, in wet compression nozzles are places compressor inlet and more water droplets are allow to entre in compressor. During compression temperature of compressed air increases in each stage and with temperature water holding capacity of compressed also increase so excess water droplets are absorbed in successive stages.

In wet compression net output of gas turbine increase so this system also used in commercial and military airplanes to get extra thrust during takeoff.

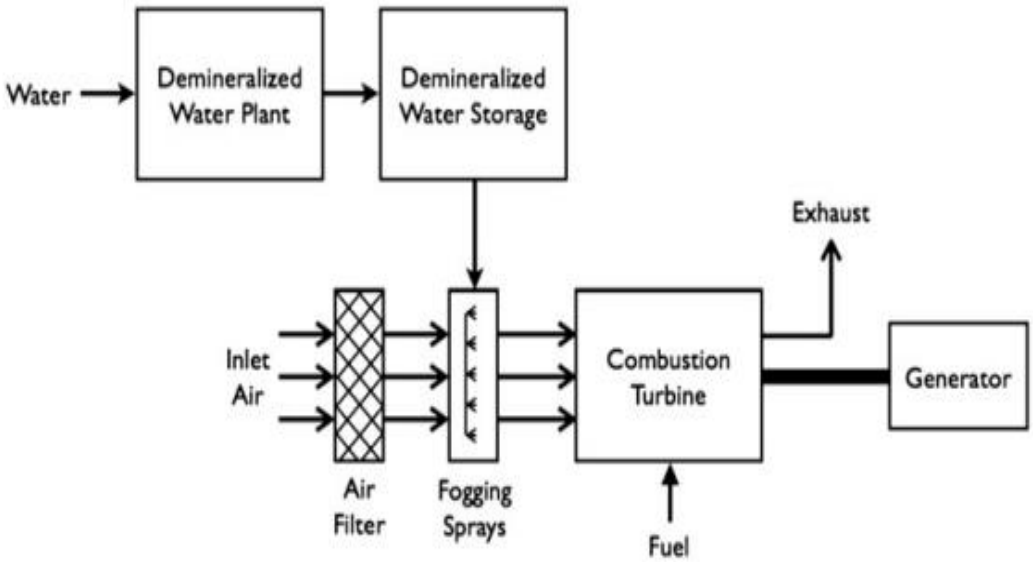

Figure 1.3 Turbine inlet air cooling with wet compression/over spray [2]

1.4.4 ABSORPTION CHILLING COOLING

Absorption chillers use heat as the energy source, instead of using electricity. This heat energy source is mainly waste steam getting from a combined cycle or hot water from a cogeneration plant, and it is allow to drive the cooling system. Compared to mechanical chillers, absorption chillers have a very low Coefficient Of Performance (COP), however, it should be taken into consideration that these cooling chillers normally uses waste heat, which decreases the operational cost and which is plus point for this system.

In absorption chilling cooling , cooling is done by vapor absorption system in which mainly of lithium bromide - water 0r water - ammonia pair are used in each pair one play role of refrigerant and other one play role of absorbent. This system work on low grade energy or waste energy like energy in turbine exhaust, solar energy and another benefit of using this system is that the minimum temperature is not constraint by ambient condition like in evaporative cooling system although minimum temperature which can be obtain is constraint by formation of ice crystal at inlet of compressor, formation of ice crystal badly damage the blade of compressor.

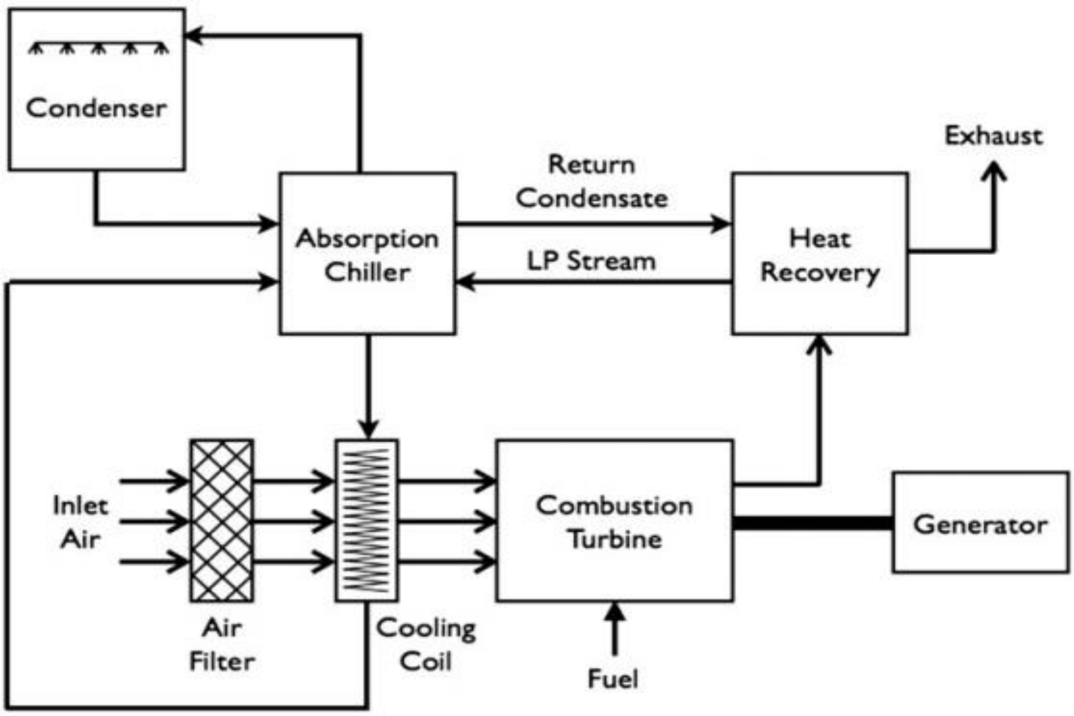

Figure 1.4 Turbine inlet air cooling using absorption chiller [2]

1.4.5 MECHANICAL REFRIGERATIVE COOLING

Mechanical refrigerative cooling based on a VCRS refrigeration cycle, in which water is cooled to a low temperature and pumped to a cooling coil installed within the turbine's filter house. In the cooling coil, chilled water absorbs the heat present in the air, cooling it to the desired temperature.

One of the principal benefits of an inlet air chilling system when compared to an evaporative cooling system is that the chilling system is not limited to cooling the air to the prevailing wet bulb temperature. The ability of the TIAC system to cool the air well below the wet bulb temperature results in significant increased power production. This is of particular importance in hot, humid climate where evaporative cooling is quite ineffective. Mechanical vapor compression TIAC systems are also capable of dry operation with no water consumption. In such environments, evaporative cooling could be technically appealing, but the large quantities of fresh water required are simply not often available.

In mechanical refrigerative cooling system , vapour compression refrigeration system (VCRS) is used to cool the inlet air. Chilled capacity of such system is measure in refrigeration ton(TR) and 1 TR capacity means removing heat at rate of 12000 Btu/hr. Compressor is major power consuming unit in VCRS which can be run by electricity, natural gas, steam turbine. Working of this cooling system is same as simple VCRS cycle, the only difference is that here inlet air is cooled in evaporator.

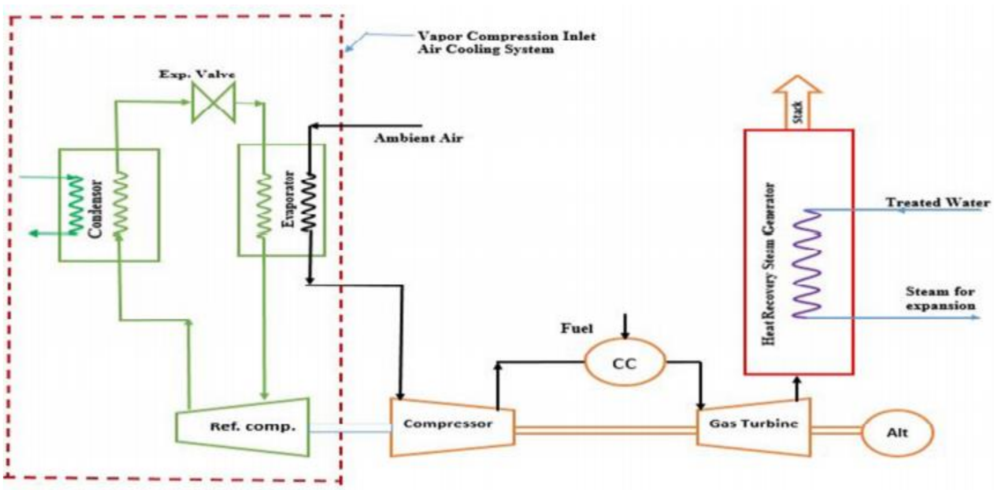

Figure 1.5 Turbine inlet air cooling system using mechanical refrigerative cooling.[2]

1.4.6 THERMAL ENERGY STORAGE

Turbine inlet cooling system with thermal energy storage use one extra component which is storage tank compare to other heat transfer system (vapor compression inlet cooling system, vapor absorption inlet cooling system). In thermal energy storage inlet air cooling system , cooling system is decoupled to cool inlet air during off peak load time and used to cool liquid in case of sensible heat thermal energy storage or to form ice in case of latent heat thermal energy storage and use heat from these thermal energy storage tank during peak load.

TES Tank is a thermal collector that allows the storage of chilled water or ice produced during off-peak time. This energy is later used during on-peak time to chill the turbine inlet air and increment its power output. A TES tank reduces refrigerant plant capacity and operational cost, producing chilled water when demand is very low, which usually coincides with the night, when ambient temperature is low and chillers have better performance. The excess of power generation will be use during the day.

Thermal Energy Storage (TES) has two operational phases: charging and discharging. During the discharge phase, the warmer and less dense returning water floats on top of the stored chilled water. The water from storage is supplied and withdrawn in low velocity, in essentially horizontal flow, so that buoyancy forces dominate inertial effects. When the stratified storage tank is charging, chilled supply water, enters through the diffuser at the bottom of the tank, and return water exits to the chiller unit through the diffuser at the top of the tank.

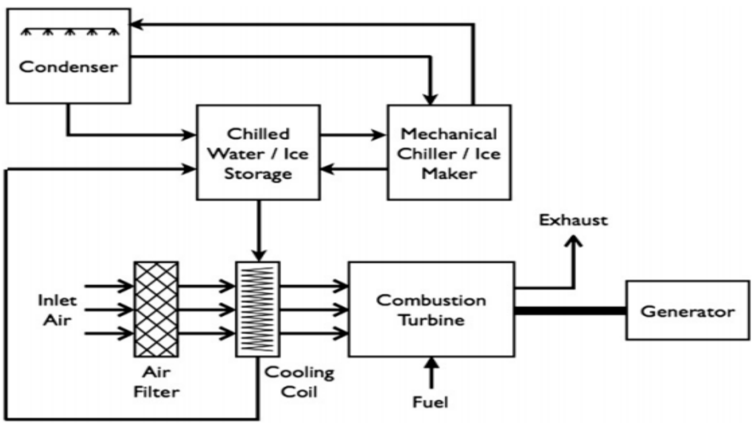

Figure 1.6 Turbine inlet air cooling system using energy storage tank[2]

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

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.[2]

In 1992 MacCracken gives 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 turbine inlet air cooling system used. [3]

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

In 1996 Maurizio De Lucia and Carlo Lanfranchi study the benefits of using different compressor inlet air cooling system for 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.[5]

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

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

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

In 2011 a study by Kim Hoon Kyounh and Kim Kyoungjin study 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.[8]

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

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

In 2015 Giovanna and Antonio Perdichizzi did techno-economic study for gas turbine inlet air cooling for combined cycle power plant for different climate condition and used cold water thermal storage as 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.[11]

In 2016 Anoop Kumar Shukla and Onkar Singh publish a Research paper on performance of gas turbine with inlet evaporative cooling and steam injected in combustion chamber and this paper work as base paper for our study and our study use all equation which are use in this paper.[12]

In 2017 Ahmed Ali Abdel Rahman, and M. A. Mokheimer work om inlet air cooling with the help of solar energy and observe that peak demand of electricity is coincides with peak of solar radiation. And concluded that 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.[13]

In 2018 a comprehensive study was conducted by Anoop Kumar Shukla,*, Achintya Sharma , Meeta Sharma , Shivam Mishra on performance improvement of simple gas turbine cycle with vapor compression inlet air cooling system and find that compressor work is strong function of inlet temperature.[14]

In 2018 A comprehensive study on simple and recuperative gas turbine cycle with inlet fogging and overspray done by Mohamed Mostafa, Yehia, Eldrainy, Mohamed M. EL-Kassaby and concluded very important result related to diameter of droplets, place of nozzle, type of duct , size of ducts.[15]

CHAPTER 3

SYSTEM DISCRIPTION

3.1 INTRODUCTION

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

3.2 ASSUMPTION TAKEN FOR THERMODYNAMICS MODELING OF SYSTEM.

- Analysis is done assuming steady flow condition.
- Air is taken as working fluid which consider as ideal gas.
- Compressor used in study is axial flow compressor.
- Specific heats of working fluid are 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 combustion chamber.

3.3 EVAPORATIVE COOLING SYSTEM.

A schematic diagram of steam injected gas turbine using evaporative inlet air cooling system shown in fig3.1 First atmospheric air is sucked by evaporative cooling system and in evaporative cooling system temperature of hot ambient air is drops from dry bulb temperature to wet bulb temperature corresponding to ambient condition and allow to entre in compressor where it compress to high temperature and pressure and entre in combustion chamber where combustion of fuel is take place. In 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 gas turbine work on some higher temperature. Atmospheric gas which is cool by evaporative cooler is consider as ideal gas and fuel used in combustion chamber is natural gas whose composition is given in table 2. The pressure loss in the combustion chamber and coolant injection in turbine blade is taken as given in Table2. Mathematical modeling of components of steam injected gas turbine based power plant is carried out on the basis of mass and energy balance.

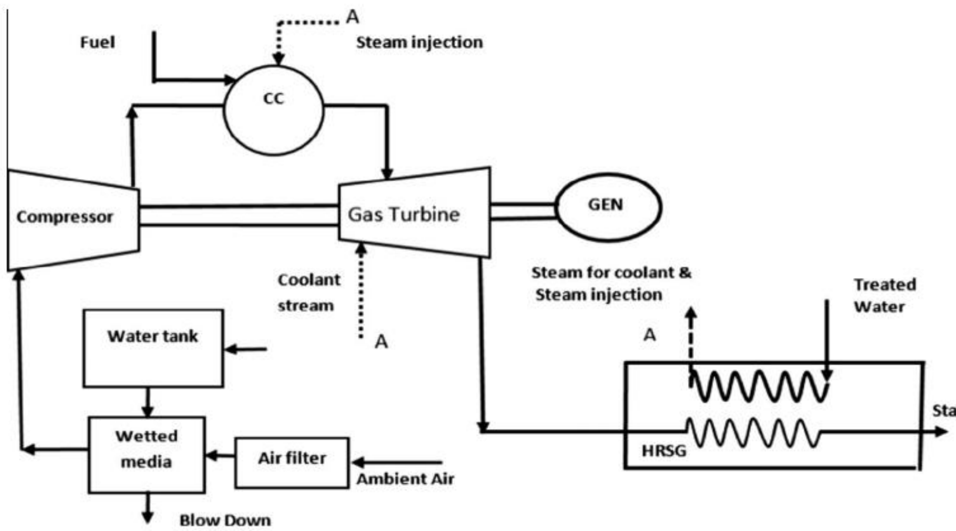

Figure 3.1 Block diagram of steam injected gas turbine with evaporative inlet air cooling.

3.3.1 Compressor

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

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

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

And

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

Where

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

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

$C_{p.a}$ 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 compressor inlet. $C_{p.a}$ and $C_{v.a}$ are only function of mean temperature along compressor which calculated as

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

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

Where T_m is mean temperature across 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 only function of average temperature along compressor.

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

$$C_{v.vap} = C_{p.vap} - 0.4614 \tag{3.8}$$

Energy and mass balance across compressor give compressor work done as

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

3.3.2 Estimation of cooling load.

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

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

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

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

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) , molar mass of water vapour(M_v) and molar mass of air(M_a) as shown in given equation:

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

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

$$Q_l = \frac{V_{amb.a}}{\upsilon_{wair}}\{\omega_{amb,DB}\left(C_{p,v}T_{DB,amb} + L\right) - \omega_{c,i}\left(C_{p,v}T_{com,i} + L\right) - (\omega_{amb,DB} - \omega_{c,i})C_{p,w}T_{com,i}\} \tag{3.13}$$

Where $C_{p,v}$, L, $\omega_{c,i}$ and $C_{p,w}$ 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.

3.3.3 Evaporative cooling

In evaporative cooling, cooling is based on the fact that evaporation cause cooling effect so when hot ambient air come 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 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.

Evaporative process is assuming to be adiabatic and temperature of water veing evaporated is equal to $T_{a,o}$, after evaporation specific humidity of air at exit of 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 mass flow rate of water required for evaporation

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

3.3.4 Combustion chamber

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

From simple mass and energy balance following equation are obtain:

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

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

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

Energy balance

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

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

Fuel air ratio

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

where SAR is steam to air ratio

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

3.3.5 Cooled gas turbine blade

Gas turbine blade cooling is used to increase the working temperature of gas turbine and improve the creep resistance, fatigue resistance and working life time. Here gas turbine blade are cooled by film cooling in which steam is allow to flow over turbine surface from one or more location where higher is achieve which is leading edge of turbine blade and some steam also injected into downstream of blade. By using the turbine blade cooling working temperature of gas turbine increase but it increase the cost of 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:

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

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.

$$\lambda = 2c/s * \cos \propto \qquad (3.23)$$

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

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

$$\varepsilon_{aw} = \frac{T_g - T_{aw}}{T_g - T_{cl,e}} \qquad (3.24)$$

Average Stanton number (St_g)

$$St_g = \frac{Nu_g}{Re_g Pr_g} \qquad (3.25)$$

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

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

3.3.6 Gas turbine

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

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

Net specific work output(w_{net}) $\qquad w_{net} = w_{gt} - w_{com} \qquad (3.28)$

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

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

Table 2

Input parameters use in evaporative cooling system analysis.

Parameter	Symbol	Unit
Compressor	(a) Polytropic efficiency ($\eta_{pt.com}$) = 92	%
	(b) Mechanical efficiency ($\eta_{m.com}$) = 98.5	%
Gas turbine	(a) Turbine blade temp. (Tb) = 1123	K
	(b) Exhaust pressure = 0.109	MPa
	(c) Exhaust hood temperature loss = 5	K
	(d) Polytropic efficiency ($\eta_{pt.gt}$) = 92	%
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) Adiabatic wall effectiveness (ε_{aw}) = 40	%
	(c) Prandtl number (Pr_g) = 0.7	
	(d) Reynolds number (Re_g)=$1*10^6$	
	(e) Lamda (λ) = 10	

3.3.7 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 evaporative cooling system for gas turbine. A program is made on EES software which consider all the equation and assumption used in base paper and results are computed based on program which are compare with the results of base paper to validate program.

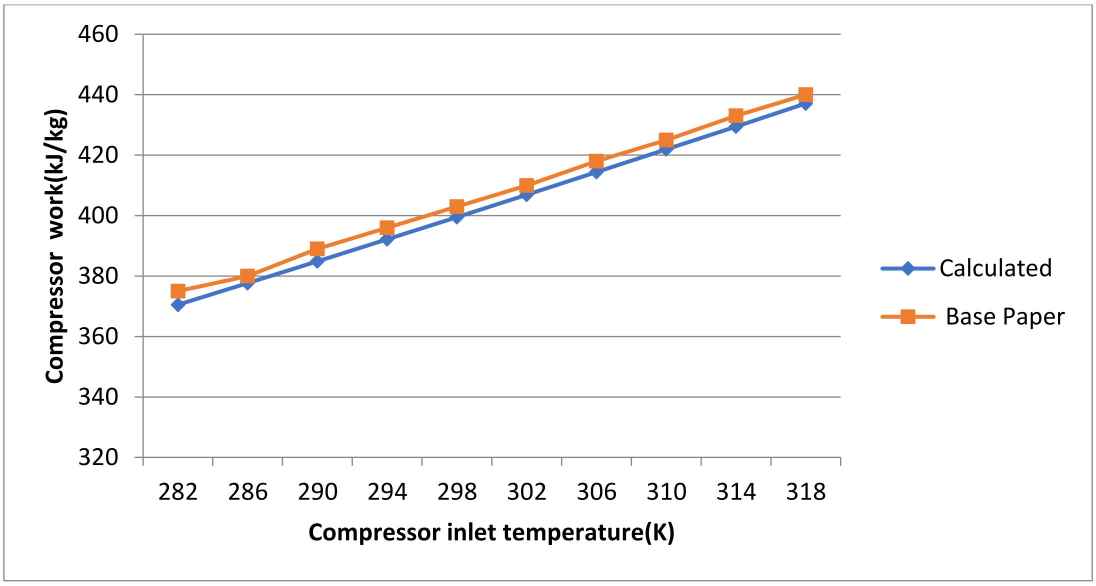

Figure 3.2 Variation of compressor work with compressor inlet temperature for system validation.

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

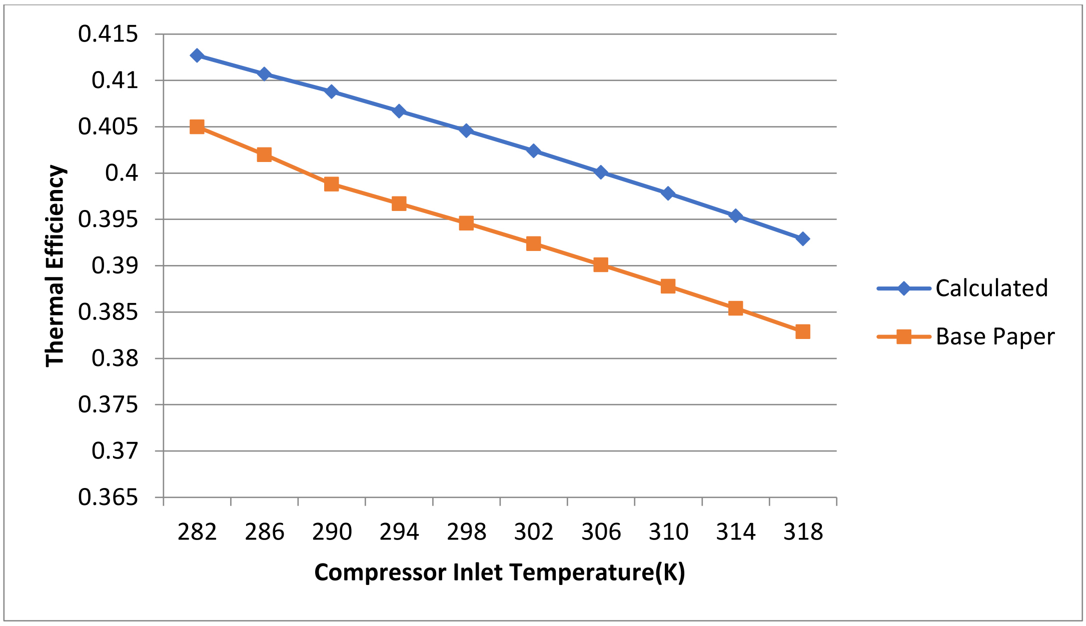

Figure 3.3 Variation of Thermal Efficiency with Compressor inlet Temperature for system validation.

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

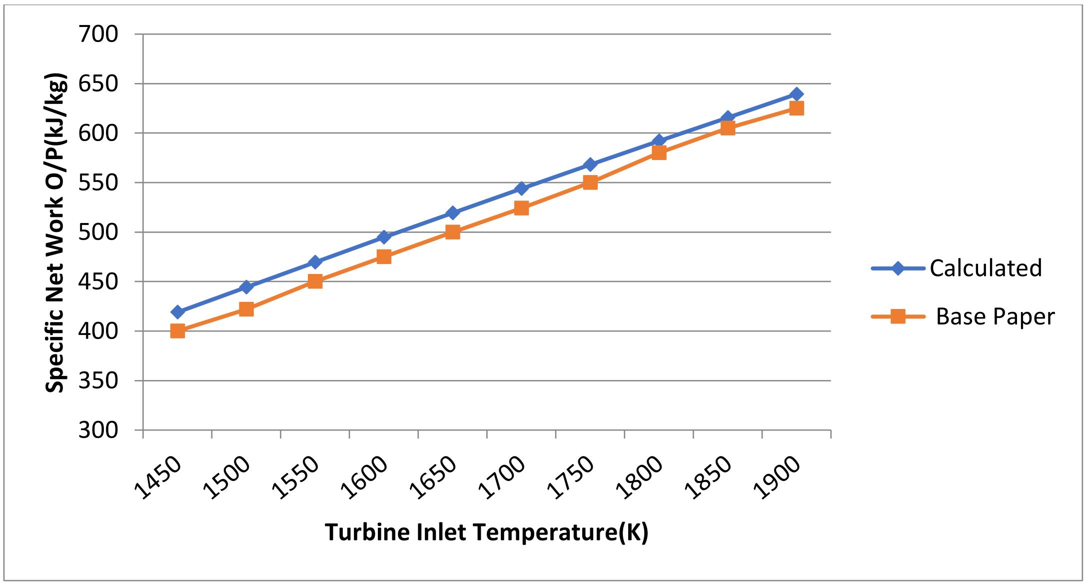

Figure 3.4 Variation of specific net work output with TIT for system validation.

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

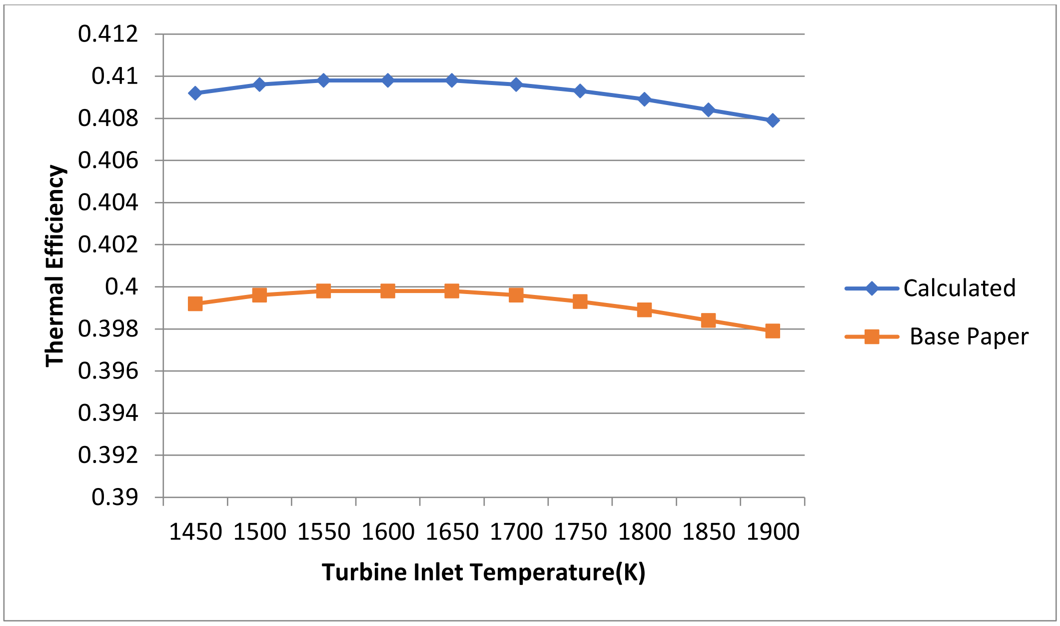

Figure 3.5 Variation of Thermal Efficiency with TIT for system validation.

Fig 3.5 depicts the decrease in thermal efficiency with 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 graph used for system validation it is observed that maximum percentage error is 4.5% which is less than 5% so it can be said that program is giving satisfactory results which are closed to base paper results.

3.4 VCRS COOLING SYSTEM

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

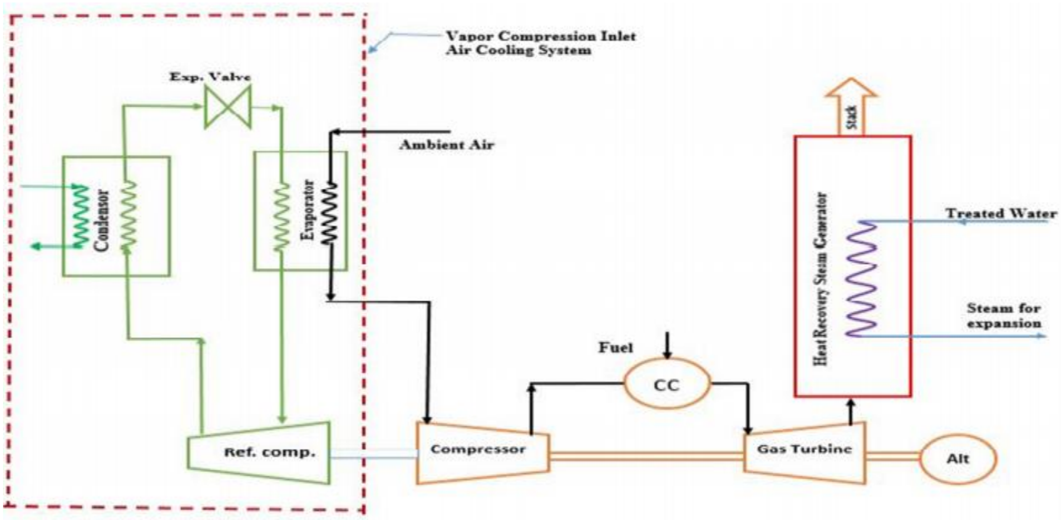

Figure 3.6 Block diagram of gas turbine with mechanical inlet air cooling system.

3.4.1 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 frige main power consuming device in this system is compressor . Hence, the actual refrigerating compression work is given as:

$$\dot{W}_{rcom} = \frac{Q_{net,cl}}{\eta_m \eta_{el} \eta_{vol}} = \frac{Q_{net,cl}}{\eta_{eu}} \quad (3.31)$$

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

$$W_{ref} = \frac{Q_{net,cl}}{COP_{vc}\eta_{eu}(1-\mu*x)^n} \quad (3.32)$$

Here µ is an empirical constant which change with the types of refrigerants and x is 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.

Total cooling load is addition of latent cooling load and sensible cooling load.

$$Q_{net,cl} = Q_s + Q_l \quad (3.33)$$

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

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

3.4.2 Compressor

Following temperature – pressure relation is used

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

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

3.4.3 Combustion Chamber

Fuel air ratio after mass and energy balance

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

3.4.4 Gas Turbine

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

$$\frac{dT}{T} = \left(\frac{dP}{P}\right)^{\eta_{pt}\left(\frac{\gamma-1}{\gamma}\right)} \tag{3.38}$$

Net specific work output(w_{net}) $w_{net} = w_{gt} - w_{com} - w_{ref}$ (3.39)

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

Table 3

Input parameter use VCRS cooling system analysis.

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

3.4.5 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 mechanical cooling system for gas turbine. A program is made on EES software which consider all the equation and assumption used in base paper and results are computed based on program which are compare with the results of base paper to validate program.

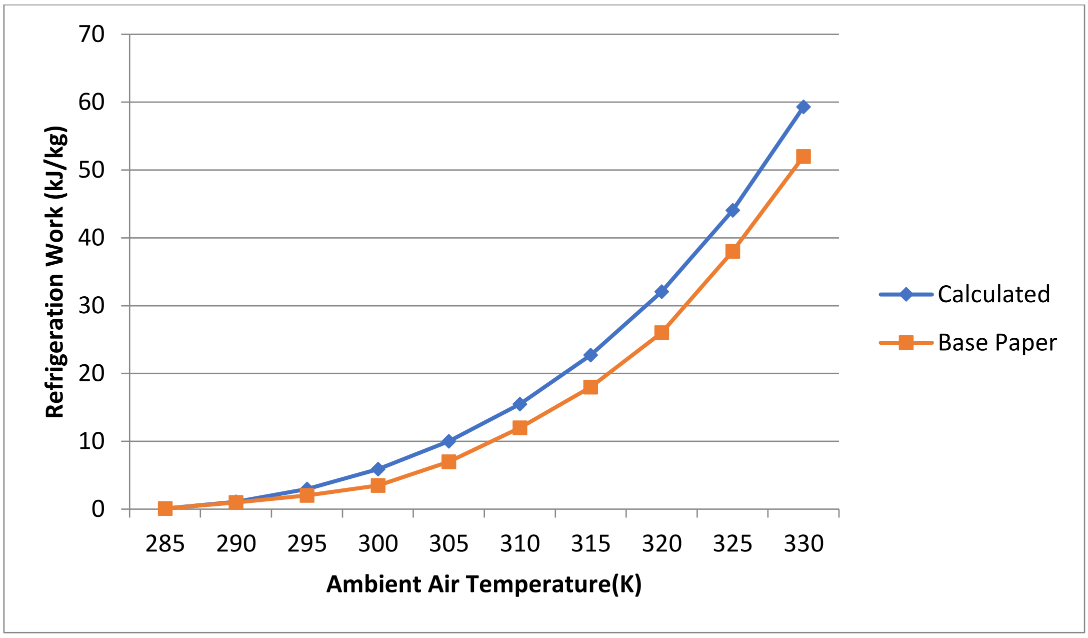

Figure 3.7 Variation of Refrigeration Work with Ambient Temperature for system validation.

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

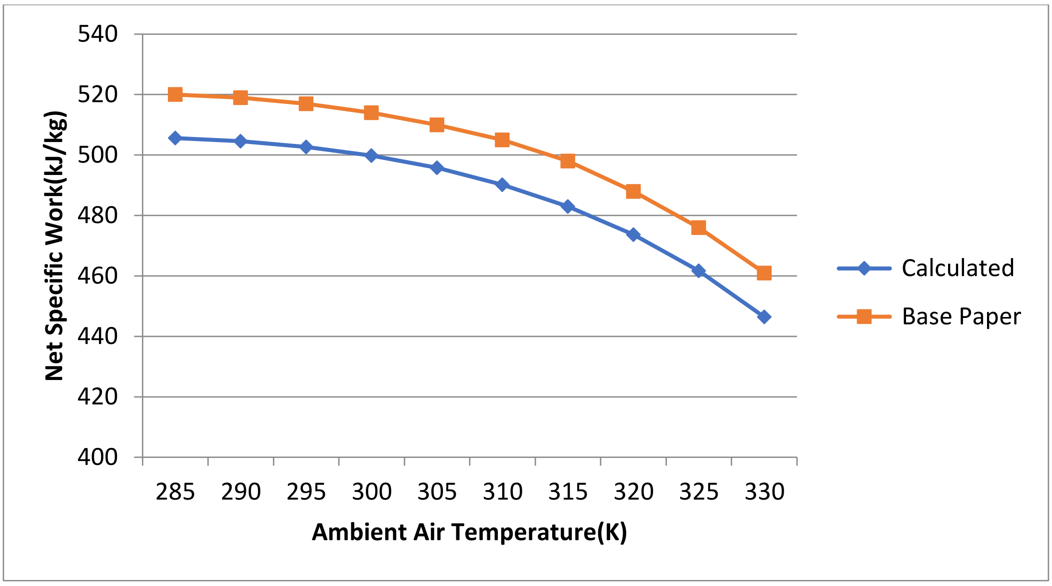

Figure 3.8 Variation of Net Specific Work with Ambient Temperature.

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

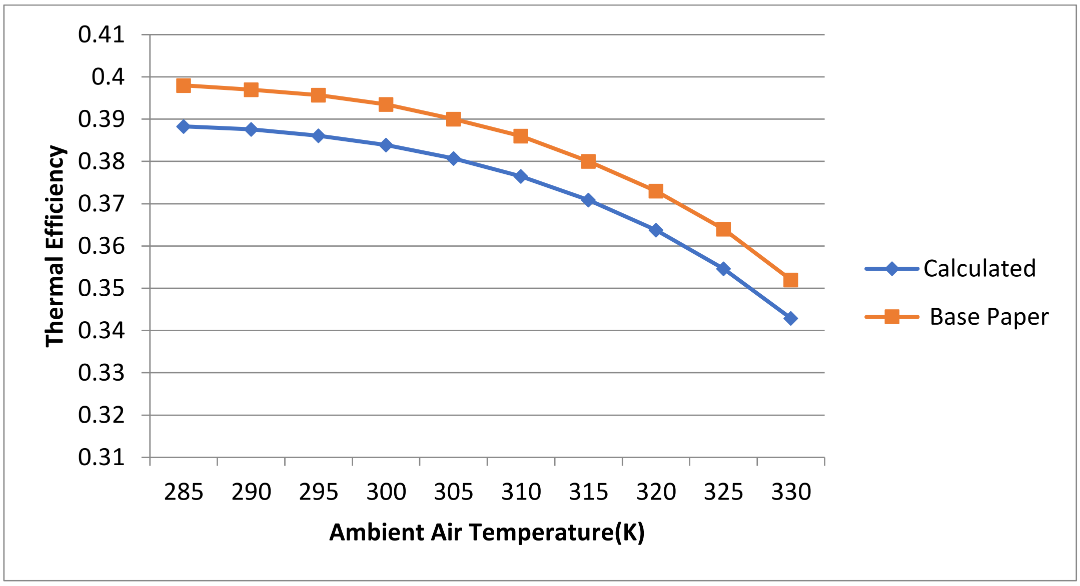

Figure 3.9 Variation of Thermal Efficiency with Ambient Temperature.

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

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

3.5 INTEGRATED COOLING SYSTEM

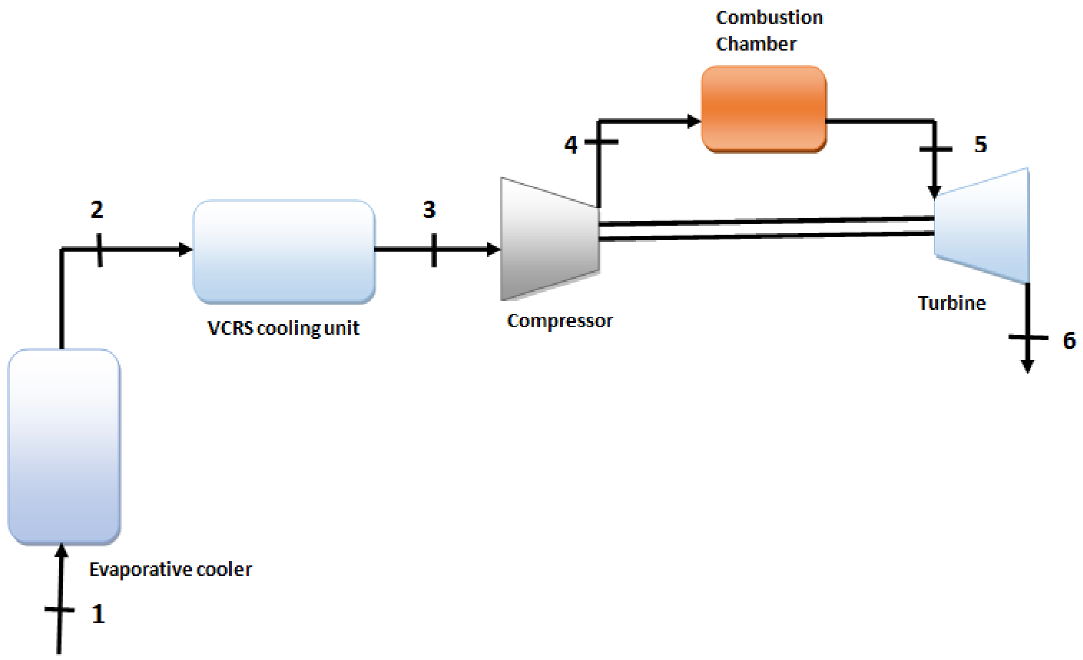

Figure 3.10 Block diagram for integrated cooling system

As seen in fig 3.10 integrated cooling system is combination of two cooling system (Evaporative cooling system and Mechanical cooling system). Main purpose of integrating two different cooling system is to use the benefits to both system 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 constrain 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 constrain by ambient air condition so any desired temperature is obtain, but capital cost of mechanical cooling system is high so to achieve a desired temperature with low capital cost both system used as integrated cooling system.

In integrated cooling system air is enter in evaporative cooling system from ambient condition where heat is transfer from hot ambient air to flowing water and ambient air cool down wet bulb temperature if evaporative cooling system have 100% effectiveness and near to wet bulb temperature if evaporative cooling system is not have 100% effectiveness and if desired CIT is below wet bulb temperature than air is pass through 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 is enter into gas turbine's compressor where it is compressed to high pressure and temperature and allows to enter in combustion chamber where heat is added after burning of fuel and amount of fuel to be burn is depend on desired TIT. This burn fuel and air mixture is allow to expand in turbine where work is produce to rotate generator and produce electricity.

Thermodynamics analysis of integrated cooling system is done with the help of equation used in evaporative cooling system and mechanical cooling system in 3.3 and 3.4 with all assumption and input parameter describe in Table 2 &3 and obtains results are explain 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, ambient temperature on gas turbine performance.

4.1 VARIATION OF DIFFERENT PARAMETER WITH AMBIENT TEMPERATURE.

4.1.1 Variation of specific net work output with ambient temperature.

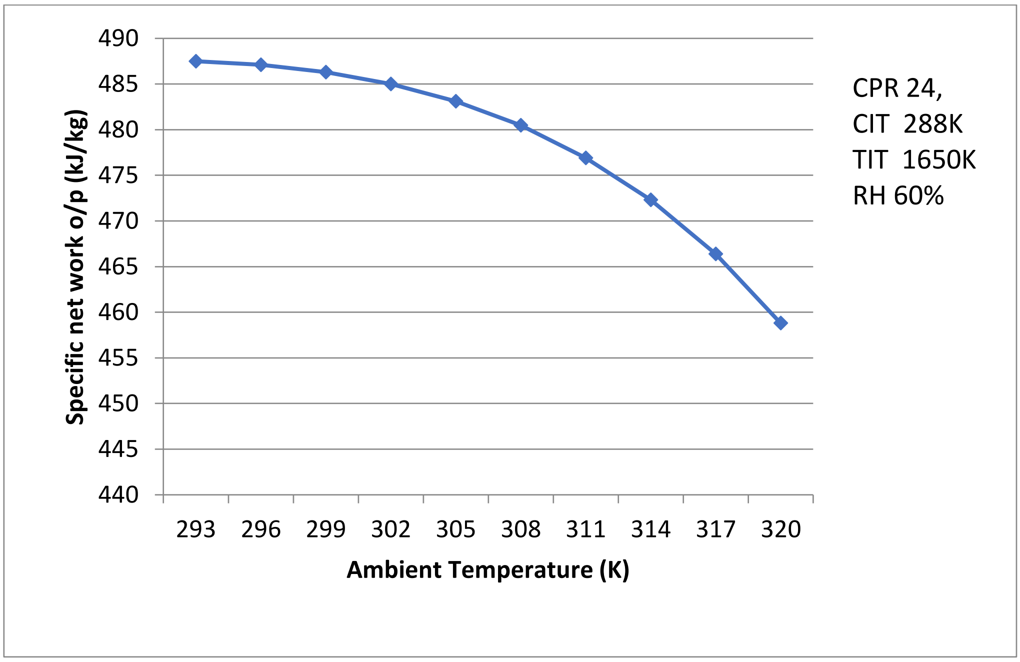

Figure 4.1 Variation of specific net work output with ambient temperature.

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 is decreases with increase in ambient temperature because net specific work output is algebraic sum of turbine work , compressor work and compressor work for VCRS unit so when ambient temperature is increases then 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 drop of 28.7 kJ/kg in net specific work done when temperature increase from 293K to 320K.

4.1.2 Variation of total cooling load with ambient temperature.

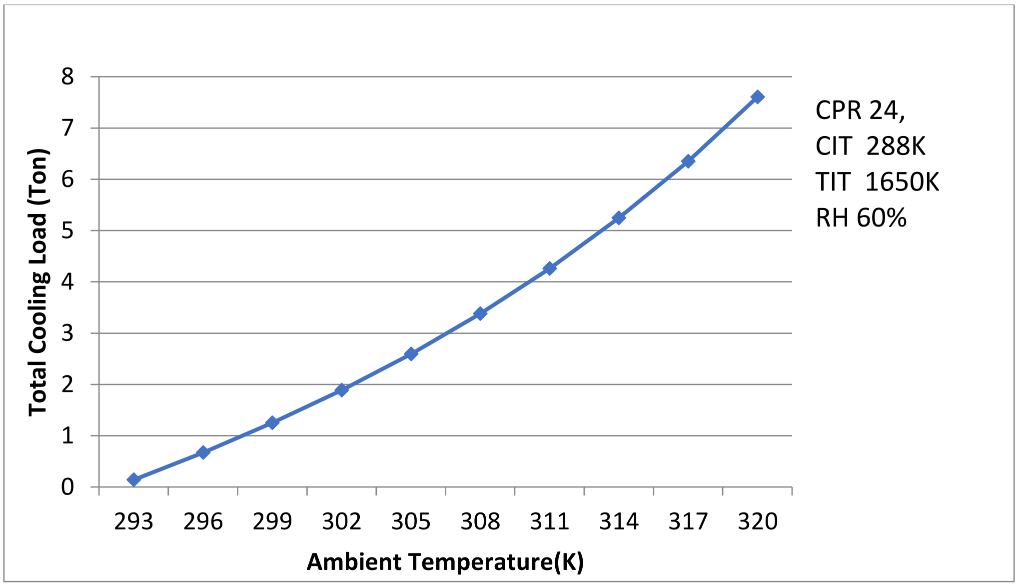

Figure 4.2 Variation of total cooling load with ambient temperature.

From Fig 4.2 it can depicts that total cooling load is 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 explain 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 increase in ambient temperature then there is increase in inlet temperature of VCRS unit which finally increase the cooling load of VCRS unit. . Cooing load increase from 1.6 kJ/s to 91.25 kJ/s when temperature increase from 293K to 320K.

4.1.3 Variation of Thermal efficiency with Ambient Temperature.

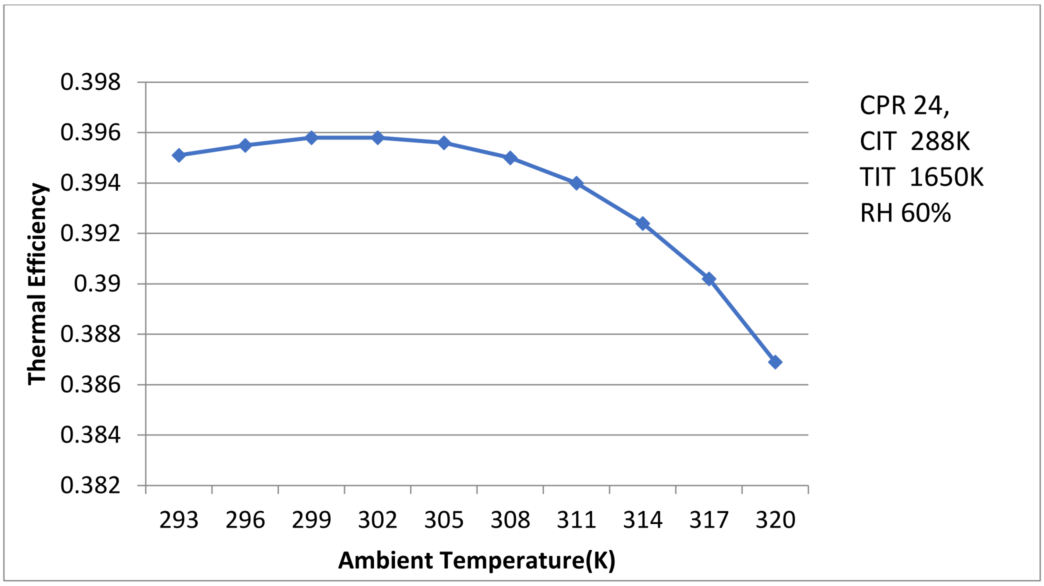

Figure 4.3 Variation of Thermal efficiency with Ambient Temperature.

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 is 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%.

4.1.4 Variation of cooling load with Ambient Temperature.

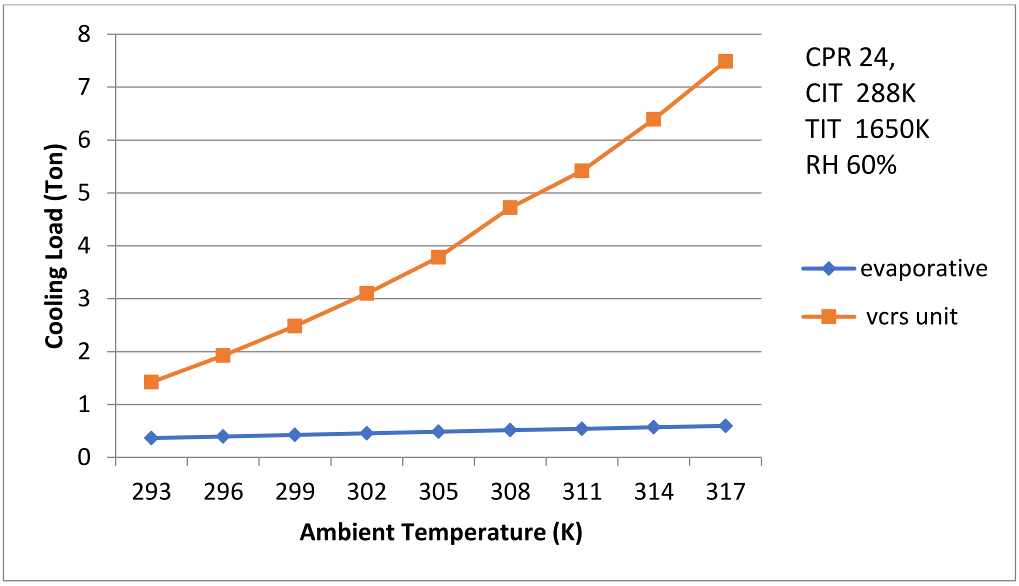

Figure 4.4 Variation of cooling load Ambient Temperature.

Fig 4.4 depicts the cooling load share by two different inlet cooling system when ambient temperature is increases. From the graph it is clear that total cooling load increases with increase in ambient temperature, at lower temperature of 293K evaporative cooling system have 4.373 kJ/s of cooling load and VCRS unit have 17.07 kJ/s of cooling load which is 79% of total cooling load and at higher temperature of 317K evaporative cooling system have 7.14 kJ/s of cooling load and VCRS unit have 89.86Kj/s of cooling load which is 92% of total cooling load i.e. with 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.

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

4.2.1 Variation of net specific work output with Turbine Inlet Temperature.

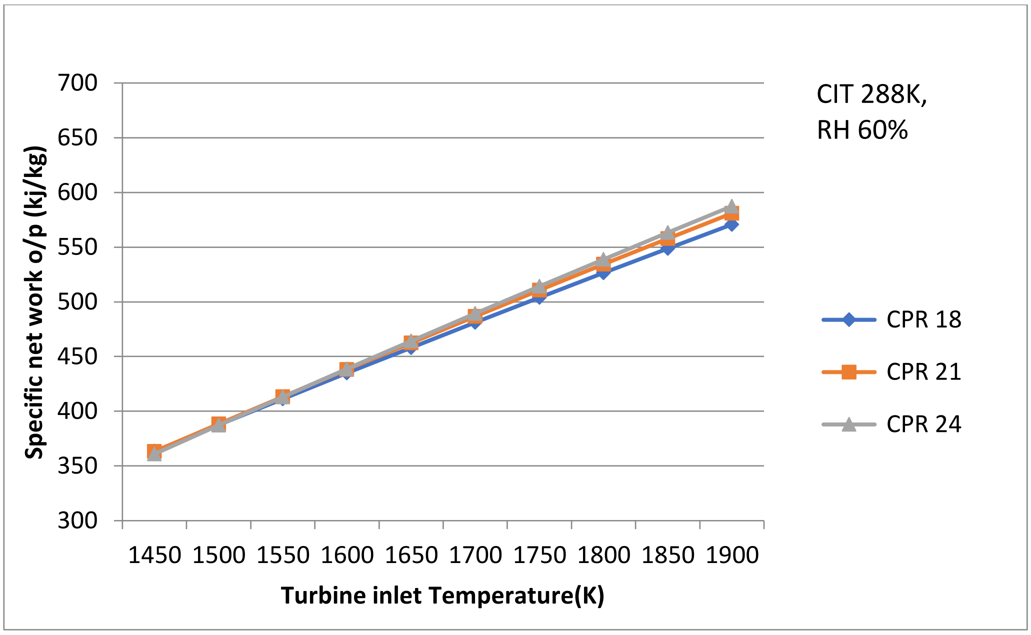

Figure 4.5 Variation of specific net work output with Turbine inlet Temperature.

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 increase in turbine inlet temperature. This happen because when turbine inlet temperature increase then work 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 increase in pressure ratio so for given graph maximum net specific work output is obtain 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 59%.

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

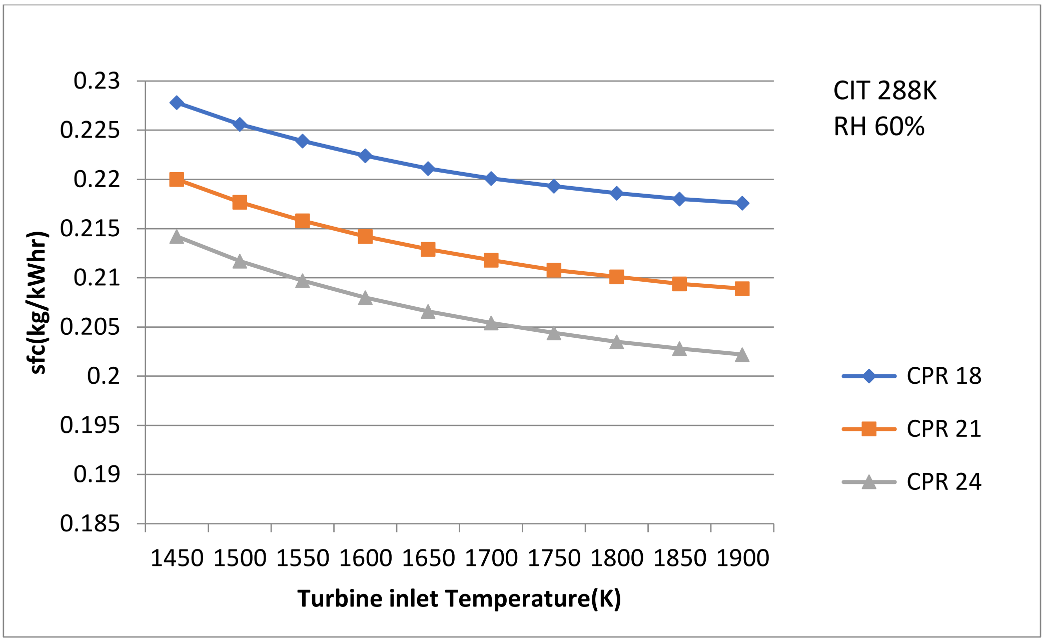

Figure 4.6 Variation of sfc with Turbine Inlet Temperature.

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

There is drop in sfc by 5% when TIT increases from 1400K to 1900K.

4.2.3 Variation of Thermal Efficiency with Turbine Inlet Temperature.

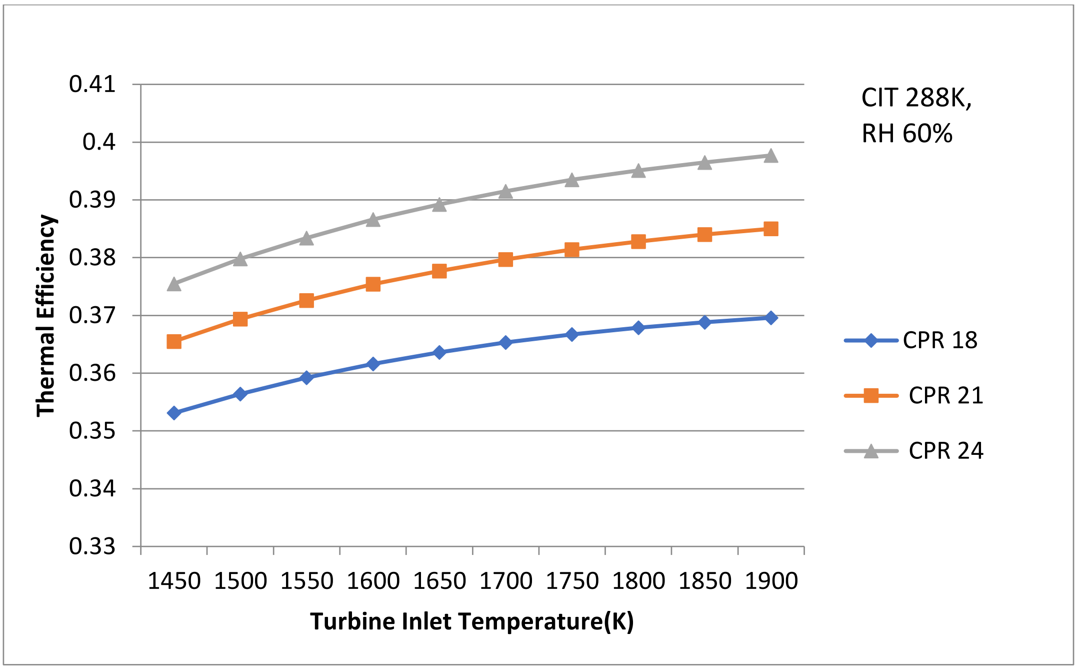

Figure 4.7 Variation of Thermal Efficiency with Turbine Inlet Temperature.

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

Thermal efficiency increase by 5% when TIT increase from 1450K to 1900

4.3 VARIATION OF DIFFERENT PARAMETRE WITH COMPRESSOR INLET TEMPERATURE (CIT).

4.3.1 Variation of compressor work with compressor inlet temperature.

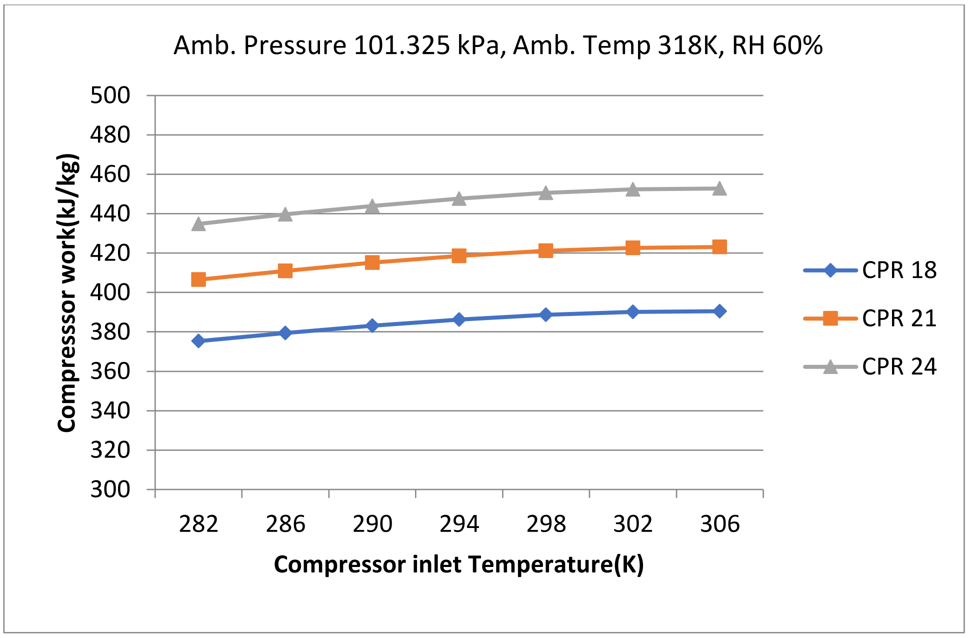

Figure 4.8 Variation of compressor work with compressor inlet temperature.

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 graph it is clearly seen that compressor work is increases with increase in compressor inlet temperature because compressor work is directly proportional to compressor inlet temperature and decrease and relative humidity with increase in compressor inlet temperature is another reason for decrease in compressor work. Compressor work also increase with increase in compression pressure ratio.

For CPR 21 when compressor inlet temperature increase from 282K to 306K compressor work increase by 4%.

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

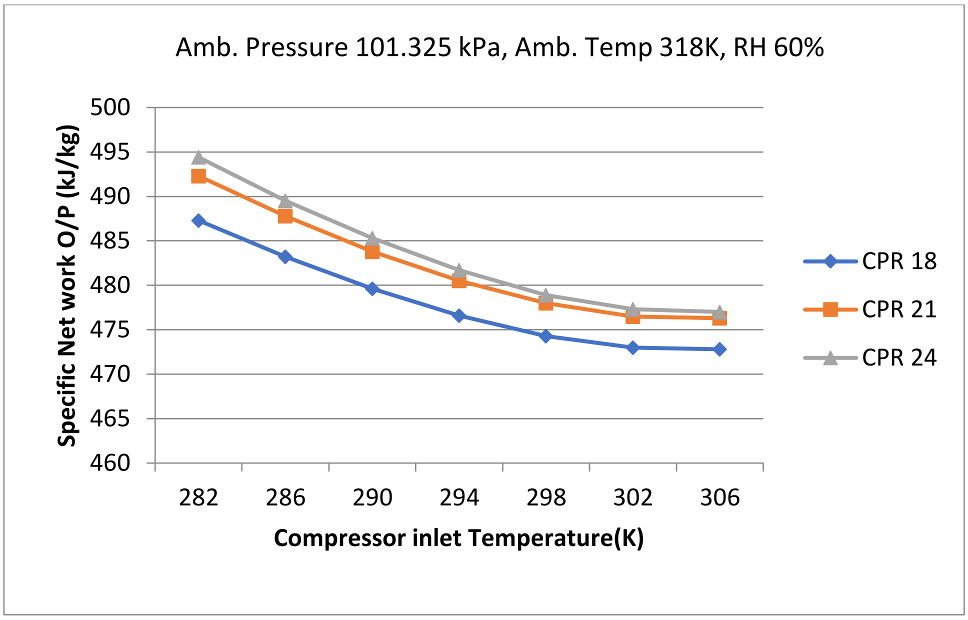

Figure 4.9 Variation of specific net work output with compressor inlet temperature.

From Fig 4.9 it can be concluded that net specific work output decrease with increase in compressor inlet temperature. Net specific work output is 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 variation of compressor work done is explain in fig 24 but compressor work done of VCRS unit is also increase 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 3% drop in net specific work when compressor inlet increase from 282K to 306K.

4.3.3 Variation of Thermal Efficiency with compressor inlet temperature.

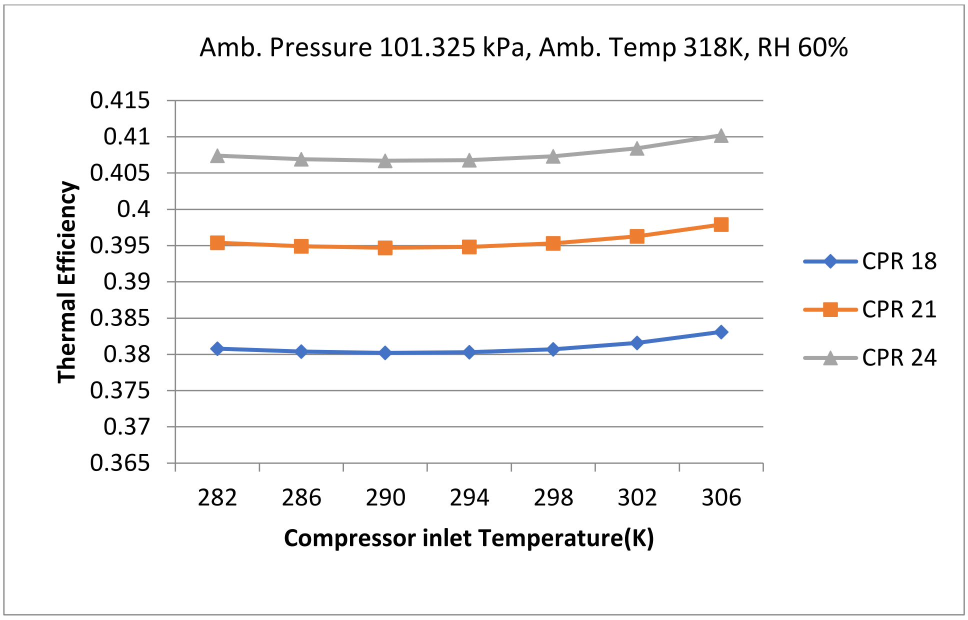

Figure 4.10 Variation of Thermal Efficiency with compressor inlet temperature.

Fig 4.10 depicts the variation of thermal efficiency with compressor inlet temperature and shows that thermal efficiency decreases with increase in compressor inlet temperature. Thermal efficiency is ratio of net specific work output and heat added in combustion chamber, for constant turbine inlet temperature heat added in 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 0.6%.

4.4 VARIATION OF DIFFERENT PARAMETRE WITH AMBIENT RELATIVE HUMIDITY

4.4.1 Variation of Net Specific Work Output with Relative Humidity.

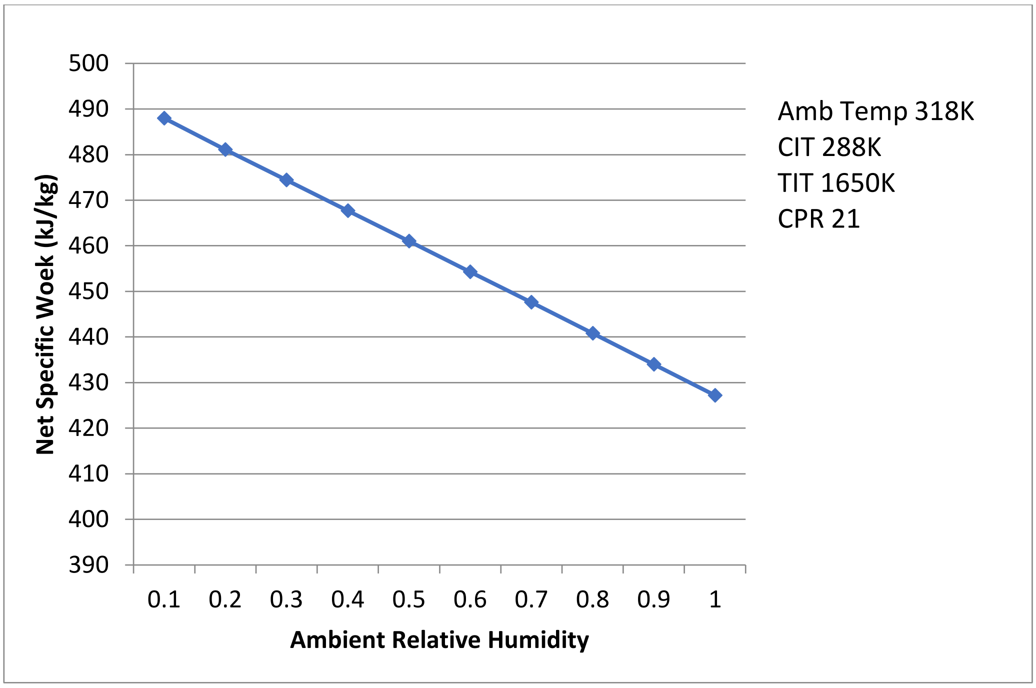

Figure 4.11 Variation of net specific work output with ambient relative humidity.

Fig 4.11 depicts the variation of specific net work output with ambient relative humidity and tells that net specific work output decreases with increase in ambient relative humidity this happens because when ambient relative humidity increases then wet bulb temperature also increases which means 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 0.1 to 1 there is drop of 60.8 kJ/kg in net specific work done.

4.4.2 Variation of Thermal Efficiency with Ambient Relative Humidity

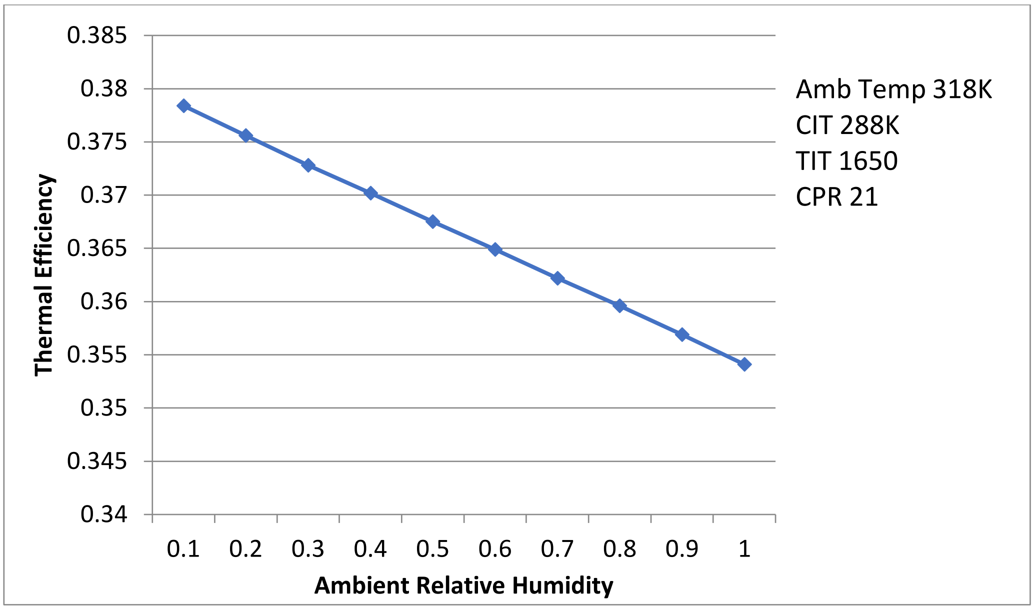

Figure 4.12 Variation of Thermal efficiency with Ambient Relative Humidity

Fig 4.12 depicts the variation of thermal efficiency with ambient relative humidity and tells that thermal efficiency decreases with increase in ambient relative humidity this happens because when ambient relative humidity increases then wet bulb temperature also increases which means 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 combustion chamber, for particular TIT heat added in combustion chamber is constant so for this case thermal efficiency is directly proportional to net specific work done so with decrease in net specific work output thermal efficiency also decrease which can be seen in fig x.

When ambient relative humidity is increase from 0.1 to 1 , thermal efficiency decrease from 0.37 to 0.35.

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

In this section variation of different parameter with ambient temperature is discussed for various gas turbine cooling system . 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 integrated system of Simple Gas Turbine with Inlet Fogging and Mechanical Cooling(SMC+IF+MC).

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

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

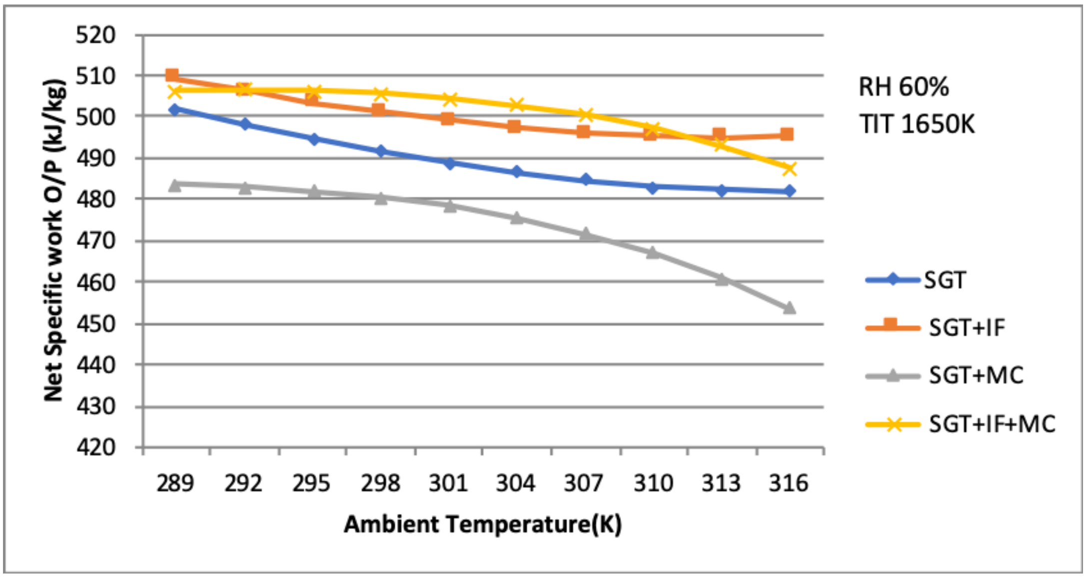

Figure 4.13 Variation of Net Specific Work Output with Ambient Temperature for gas various turbine cooling system.

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

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

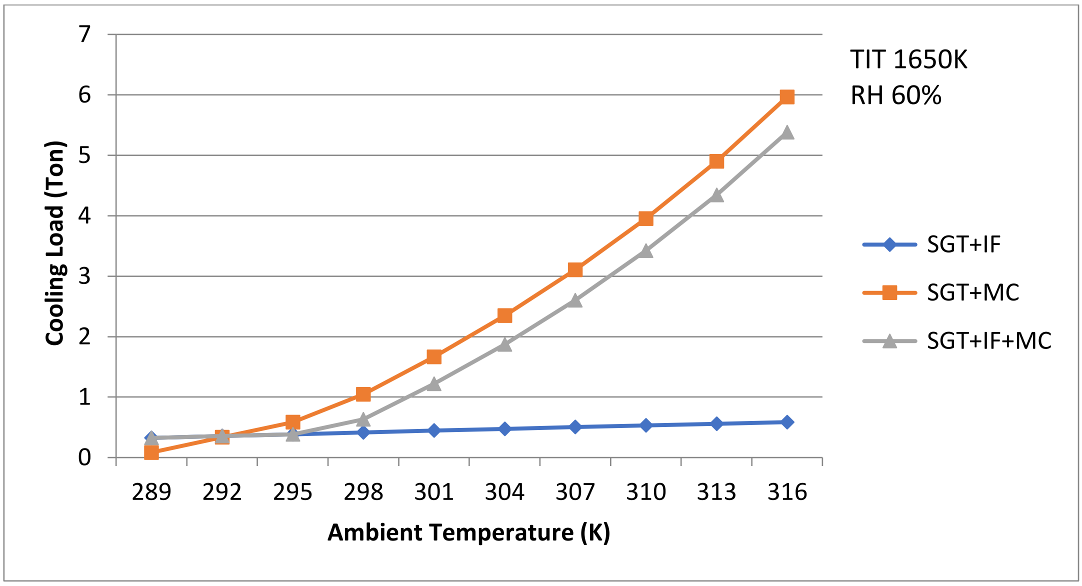

Figure 4.14 Variation of Cooling load with Ambient Temperature for various gas turbine cooling system

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

There is average difference of 8kJ/kg in cooling load of mechanical system and integrated system after 300K.

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

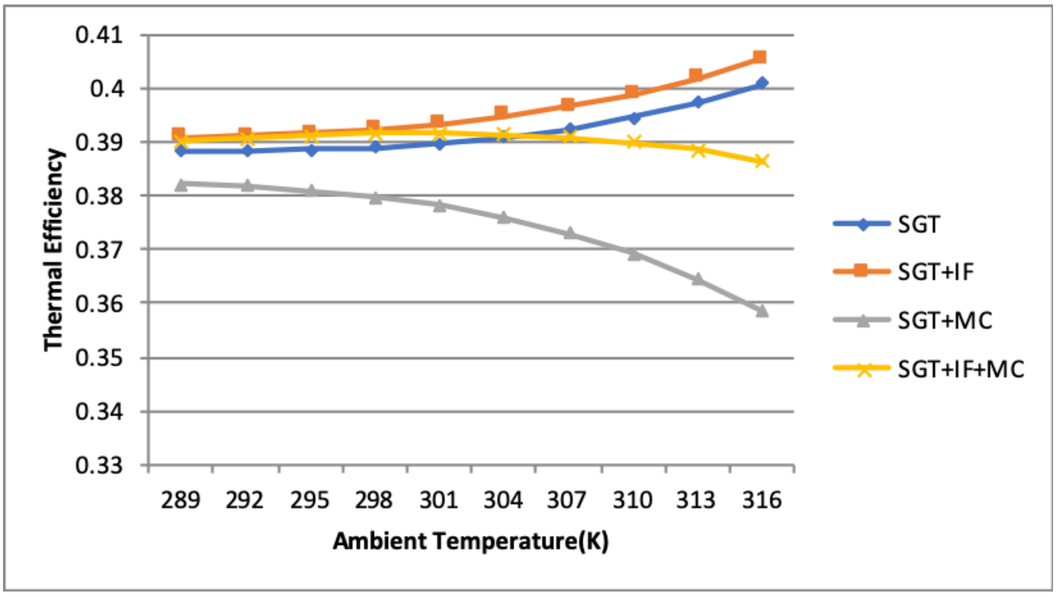

Figure 4.15 Variation of Thermal Efficiency with Ambient Temperature for various turbine cooling system.

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 various cooling system following conclusion have been drawn.

- Increase in ambient temperature degrade the net specific work and thermal efficiency of gas turbine and increase the cooling load. There is 5% decrease in net specific work output, 53% increase in cooling load and 20% decrease in thermal efficiency when ambient temperature increase from 293K to 320K.
- Gas turbine performance improve by increase in TIT. Net specific work output increase by 59%, thermal efficiency increase by 5% and SFC decrease by 5% when TIT increase from 1450K to 1900K at CPR 21.
- Compressor work increases with increase in CIT by 4% at CPR of 21 and efficiency increases by 0.4 % with decrease in net specific work output by 3.2%.
- When relative humidity increase from 1% to 100% net specific work decrease by 12% and thermal efficiency decrease by 6%.
- By comparing various cooling system with integrated cooling system it can be conclude that integrated cooling system give maximum net specific work O/P with minimum cooling load and moderate efficiency.
- From all the result it can be concluded that integrated cooling system is one of the best cooling system for hot and dry region.

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