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Augmenting Gas Turbine Performance through Inlet Air Cooling

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ABSTRACT. The improvement in gas turbine power plants by integrating a mechanical chiller to it for cooling the inlet air is the prime objective of this study. The bucket cooling method adopted in this study to cool the turbine buckets has been chosen as film cooling. The selected input parameters have been varied to analyze the power plant output and efficiency at varying conditions and then select the best alternative which will help the design engineers. The integration of mechanical chiller significantly augments the plant output and efficiency. This improvement is more pronounced in hot and humid climates. It was observed that as the compressor inlet temperature is reduced the bucket coolant mass flow rate decreases and the mass of fuel energy input increases. The increase in specific work and efficiency is calculated to be 14.59 % and 4.46 % respectively when the ambient temperature drops to 283K. The work ratio increases with increase in value of ratio of inlet temperatures (r_{IT}) upto 5.6 after which it starts decreasing. There is an optimum r_{IT} at any pressure ratio ($r_{p,c.}$)at which the work ratio is maximum . The heat rate increases with increase in r_{IT} and decrease in $r_{p,c.}$.
Keywords: Mechanical Chiller; film blade cooling; gas turbine performance; inlet-air cooling; ambient;

I. INTRODUCTION

One of the major parameter that effects the performance of gas turbine plant is inlet air cooling. As the inlet air temperature increases not only that we suffer a reduction in gas turbine performance but also the pollutants increases. Since the benefit of reducing the inlet air cooling is multifold, the technique has been widely adopted in the power plant industries and is further investigated in this study. The most significant and well-known technique to cool the inlet air of the gas turbine is cooling with: absorption chillers, Mechanical chillers and evaporation systems

Gord and Dashtebayaz [1], suggested that the performance and efficiency of gas turbine can be improved by using turbo-charger. In that approach, a comparative study between a common air cooling and a turbo-air cooling method using turbo-expanders has been portrayed. The mechanical efficiency of a plant can be increased using a mechanical Chiller where as efficiency in electricity production enhanced significantly high by using turbo-expander as reported. According to Popliet. al [2] , an absorption cooling system integrated with a conventional evaporative coolers and mechanical vapor compression chillers powered by waste heat produces better power output. A gas turbine of evaporative cooling system , power output and energy efficiency has been enhanced by 4.2% and 1.6% respectively. In comparison, in a vapor absorption cooling, an increase of 23.2% by power output and 13% by energy efficiency has been reported. However, vapor compression cooling established an annual saving of 2MW electric power.

Gas turbines used in plants, mostly faced an adverse effect in power output when ambient temperature increased. It has been reported that gas turbine power generation declined 15% of the rated power when the ambient temperature increased from 15^0 C to 36^0 C [3]. In order to compensate such power loss, new technology has been introduced. Inlet air cooling is regarded as a promising method for improving gas turbine efficiency up to 30% at one third cost of new turbine and half that of a peaking plant.

Inlet air cooling techniques to gas turbine during combined cycle was addressed a subject matter to improve plant efficiency. Najjar et al[4] investigated on inlet air chilling effect in gas turbine by introducing a cooling coil and reported that the turbine output improved by 10% and 18% during cold humid conditions and hot humid conditions respectively. Amell and Cadavid [5] have discussed the effect of the relative humidity on the atmospheric air-cooling thermal load, for gas powered thermal station installed in Colombia, when implementing cooling techniques such as: vapor compression and ice storage. G Srivastava and R Yadav [6] investigated collaboratively on the influence of relative humidity on the performance of a combined cycle using vapor compression refrigeration system. It has been reported that cooling the inlet air by 20 K using vapor compression refrigeration system, improves the plant specific work and plant efficiency by around 4% and 0.39% respectively. Lucia et al. [7] have investigated on air cooling system and portrayed a comparative study of Cogeneration of gas turbine power plant and concluded that in the Italian climate, the turbine power output may increase by 18% to 19% if the compressor inlet air is cooled to 10°C. This may be due to reducing the loss of energy due to excess heating. Al-Ansari and Ali [8] discussed on a hybrid turbine inlet air cooling (TIAC) system and reported that there was an enhancement of more than 10% of power. Sanaye et al. [9] have performed a thermo-economic analysis of ice thermal energy storage system for gas turbine inlet cooling application. The addition of inlet air cooling has been reported to enhance power by 3.9 % to 25.7 % and efficiency by 2.1% to5.2% while the payback period was increased by 3.7 years.

Another aspect of increasing the performance of gas turbine power plant by increasing the turbine inlet temperature (TIT), which has been addressed in addition to cooling of the inlet air. Such improvements in achieving the active cooling become possible by maintaining the temperature of the various components of turbine exposed to hot gases lower by continuous bleeding of fresh air from air. In addition, such high pressured bleeding of air extends the life period of components by controlling the unfavorable combined exposure to oxidation, creep, and thermal stresses on them. Bucket cooling is also another sustainable method of cooling by bleeding inlet air from compressor flowing through the internal passages in the buckets. Extensive works in this area have also been reported by many researchers [10-23]. Horlock et al. have discussed the consequence of increased energy losses associated with the cooling flow rates required due to increase in efficiency at higher temperature of inlet air. In addition, high temperature of inlet air associated with the turbine increase excess NO_X emission. In order to control the NO_x emission, excess mass flow rate of coolant and inlet air temperature of turbine, necessary control mechanism by the integration of inlet air cooling systems to gas turbines cycle plants must be employed .

A little work has been reported on thermodynamic analysis of gas turbines and combined cycle plants employing combined mode of inlet air cooling and gas turbine bucket cooling. As discussed by mohaptra and sanjay [25,27], plant efficiency and plant specific work has been improved by 2.29% and 13.9% respectively by introducing evaporative inlet air cooling and transpiration bucket cooling. Combined effect of air humidifier and inlet air cooling to film cooled gas turbine increases the efficiency. This may be due to increased work out put at higher ambient temperature and lower relative humidity [13]. Performance on exergy and emission of a cooled GT cycle subjected to evaporative inlet air cooling have also been studied by Anupam[14]. Attempts have been taken to highlight such three methods representing substantial improve in plant efficiency by 11.3% and plant specific work by 19.7% of an evaporative inlet air cooled gas turbine cycle plant introducing an air humidifier [28]. Effect of absorption inlet air cooling on combined cycle plant performance [29] has been studied and reported as increase in optimum efficiency of gas turbine cycle by 7.48% and specific work by more than 18 %.

After extensive work done on the area of inlet air cooling, a research gap has been found. Hence the literature survey pleads for further research, which has been addressed here in;

- During analysis of combined effect of vapor compression inlet-air-cooling and air film bucket cooling on gas turbine cycle ,no where the performance has been discussed.
- There is a lack of information found while studying the factors affecting the performance parameters of air film cooled gas turbine cycle integrated with vapor compression inlet-air-cooling like ambient temperature, relative humidity, compressor pressure ratio and turbine inlet temperature.
- Incompetency in mapping performance for designing a vapor compression inlet-air cooled gas turbine cycle with film cooling of turbine buckets has been identified.
- Since performance of a gas turbine cycle is highly influenced by turbine and compressor inlet temperatures, in order to study the effect of its variation on cycle performance, the ratio of inlet temperatures need to be addressed.

Hence the prime objective of reducing the research gap has been maintained by parametric analysis of the impact of vapour compression inlet air cooling to a gas turbine cycle with air film cooling of turbine buckets.

MODELING OF GOVERNING EQUATIONS

Fig. 1(a) depicts a Mechanical Chiller based VC-IAC gas turbine configuration with film bucket cooling. Different components of the said configuration have been modelled using appropriate governing equations. Certain assumptions have also been made during the modeling of such governing equations.

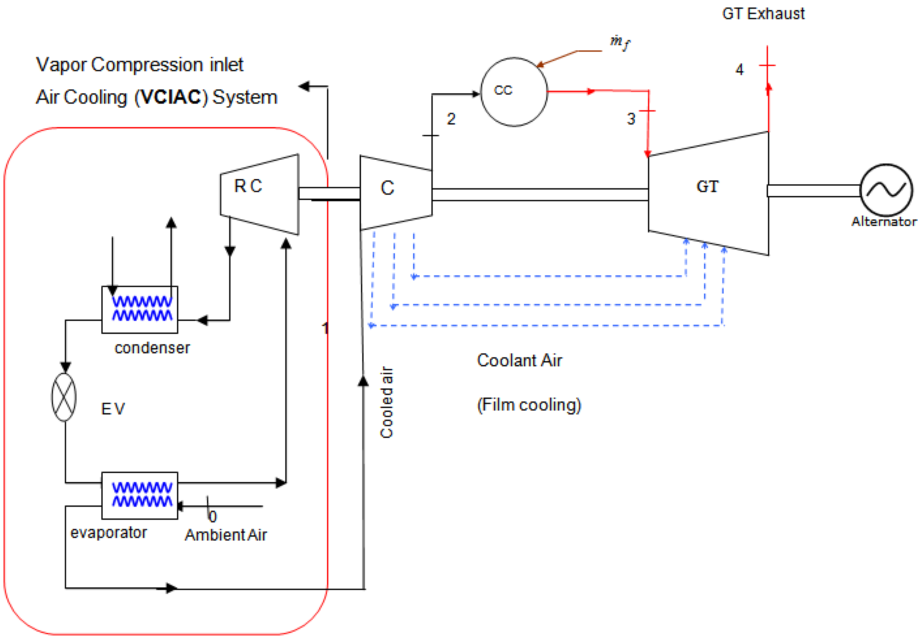

Fig 1(a). Schematic of a gas turbine configuration integrated with a mechanical chiller

Gas Model

The specific heat of gas (real)is a function of temperature and pressure. Howeverin the present study it is assumed to vary only with temperature as per the following polynomial.

$$c_{pg}(T) = f_h * \left[0.991 + \left(\frac{6.997T}{10^5}\right) + \left(\frac{2.712T^2}{10^7}\right) + \left(\frac{6.997T^3}{10^{10}}\right)\right] \quad (1)$$

In the above equation the co-efficient used in the polynomiala, b, c, and d are taken from the Touloukian and Tadash [17]. another term' f_h 'has been introduced to consider the effect of rise in specific humidity of air across the VC-IAC system such that

$$f_h = 1 + 0.05\phi_{h,e} \quad (2)$$

being the % relative humidity at the outlet of VC-IAC system.

The heat content of gas termed as enthalpy is expressed as

$$h = \int_{T_a}^{T} c_p(T)dT \quad (3)$$

It is assumed that the air entering the inlet air cooler has zero enthalpy.

Modelling of Vapor Compression Inlet Air cooling system.

The process that the ambient air undergoes inside the inlet air cooler can be represented in the psychometric chart as shown in Fig 1 (b).

As the air enters the VC-IAC and comes out at a reduced temperature, the path traced by the air in psychometric chart is as detailed in Fig. 2(a). The temperature of ambient air starts to drop due to the rejection of sensible heat and there is simultaneous rise in relative humidity till it reaches the dew point (point 'b').Any further cooling beyond this point demands the condensation of water vapor and so the quantity of heat required to be removed rises sharply until the required state at point "c" is attained.

The total amount of heat required to be removed from the ambient air for the desired state 'c' to be achieved is nothing but the summation of the sensible and latent heat of air.

Hence the refrigerating effect (total cooling load) can be calculated as under

$$Q_{CL} = \dot{m}\,(q_{sensible} + q_{latent}) \quad (4)$$

A vapor compression cycle has been adopted in the study for the refrigeration process. The work required to operate the refrigeration cycle is derived from gas turbine output.

The work of refrigeration needed for cooling the inlet air has been extracted from the gas turbine output. Several losses are to be encountered in the process of refrigeration due to which the actual power is always greater than the theoretical one.

The actual power of compression for refrigeration can be calculated as

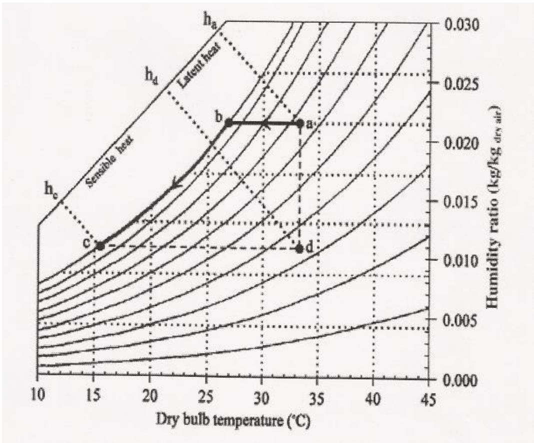

Fig. 1(b). Representation of air cooling process on psychometric chart

$$W_{comp,ref} = \frac{\dot{m}_a(h_a - h_c)}{\eta_m \eta_{el} \eta_{vol}} = \frac{\dot{m}_a(h_a - h_c)}{\eta_{eu}} \quad (5)$$

Where $\eta_{vol}, \eta_{el}, \eta_m$ and are the volumetric,electricaland mechanicalefficiency respectively.

The refrigeration work for a selected COP of vapor compression refrigeration is calculated by using the following relation

The work of refrigeration for the VC-IAC system for chosen value of COP can becomputed as [18]

$$W_{ref} = \frac{Q_{CL}}{COP_{VC}(1-\mu x)^n \eta_{eu}} \quad (6)$$

In the above formulae 'μ' is a constant which is chosen depending upon the refrigerant type and x is the quality ofair at the exit of VC-IAC. Another empirical constant 'n' is selected based on number of compression and expansion stages. In this study the value of 'μ is 0.77 (Freon 22) and the value of 'n' is 1 (single stage simple refrigeration cycle). In general the value of 'η_{eu}' is decided by the manufacturer depending upon the pressure ratio and the compressor type.

The actual COP can be computed by presenting a term refrigeration efficiency (η_{Ref}) which can be defined as

$$\eta_{ref} = \frac{COP_{actual}}{COP_{theoretical}} \quad (7)$$

- **Compressor Model**

An axial flow compressor is used in our study.Aterm polytropic efficiency ($\eta_{pt,c}$) is introduced in the equation relating pressure and temperature to account for the thermodynamic losses in the compressor as follows.

$$\frac{dT}{T} = \left[\frac{R_c}{\eta_{pt,c} c_{p,c}}\right] \frac{dp}{p} \quad (8)$$

where R_c is the gas constant such that

$$R_c = c_{p,c} - c_{v,c} \quad (9)$$

where

$$c_{p,c} = c_{p,a} + \omega_{a,i} c_{p,vap} \quad (10)$$

$$c_{p,c} = c_{p,a} + \omega_{a,i} c_{p,vap} \quad (11)$$

where $c_{p,a}$ and $c_{v,a}$ are calculated from thebelowmentioned relation [19]:

$$c_{p,a} = \frac{8.314}{28.97}(3.653 - 1.337 \times 10^{-3} T_{av} + 3.294 \times 10^{-6} T_{av}^2 - 1.913 \times 10^{-9} T_{av}^3 + 2.763 \times 10^{-13} T_{av}^4) \quad (12)$$

$$c_{v,a} = c_{p,a} - 0.287 \quad ($$

Table 1. Input data for analysis [14,15]

PARAMETER	SYMBOL	UNIT
Gas Properties :	$c_p = f(T)$ Enthalpy $h = \int c_p(T)\,dT$	kJ/kg K KJ/kg

Inlet Air Cooling System	i.Refrigerant i.Refrigeration Efficiency =45 i.Energy use efficiency (η_{eu}) =85	R22 % %
Compressor	i.Polytropic efficiency(η_{pc})=92.0 Mechanical efficiency(η_{m})=98.5 Inlet plenum loss= 0.5% of entry pr.	% % bar
Combustor	Combustor efficiency (η_{comb})=99.5 Pressure loss (p_{loss})=2.0% of entry pressure Lower heating value (LHV)= 42.0	% bar MJ/kg
Gas turbine	Polytropic efficiency (η_{pt})=92.0 Exhaust pressure=1.08 Exhaust hood loss=4 Turbine Bucket Temperature= 1123	% bar K K

Again $c_{p,vap}$ and $c_{v,vap}$ are calculated as follows[16]:

$$c_{p,vap} = \frac{8.314}{18.02}(4.07 - 1.108 \times 10^{-3}T_{av} + 4.152 \times 10^{-6}T_{av}^2 - 2{,}964 \times 10^{-9}\, T_{av}^3 + 8.07 \times 10^{-13}\, T_{av}^4) \quad (14)$$

$$c_{v,vap} = c_{p,vap} - 0.4614 \quad (15)$$

The enthalpy at any point can be calculated using the relations(1), (3), and (8).

The mass and energy balance of the compressor helps us to find out the work of compression as follows.

$$\mathbf{m_{c,i} = m_{c,e} + \sum m_{coolant,j}} \quad (16)$$

$$-\mathbf{W_c} = \dot{m}_{c,e}.h_{c,e} + \sum \dot{m}_{coolant,j}.h_{coolant} - \dot{m}_{c,i}.h_{c,} \quad (17)$$

- **Combustor Model:**

The mass and energy balance of the combustor helps us to find out the mass of fuel need to be supplied to attain the desired TIT as per following.

$$\dot{m}_e = \dot{m}_i + \dot{m}_f \quad (18)$$

$$\dot{m}_f.\Delta H_r.\eta_{comb} = \dot{m}_e.h_e + \dot{m}_i.h_i \quad (19)$$

- **Cooled Gas Turbine**

In this article, the gas turbine buckets have been cooled by film air cooling model.The mass of coolant need to be supplied to a row of buckets isexpressed as [20]:

$$\frac{\dot{m}_c}{\dot{m}_g} = \lambda \cdot \left[\frac{c_{p,g}}{c_{p,c}}\right] \cdot \left[\frac{\bar{h}_g}{c_{p,g}.\rho_g.C_g}\right] \cdot \left[\frac{(T_{g,i} - T_b)}{\varepsilon(T_b - T_{c,i})}\right] \quad (20)$$

where $S_g \cong 2c$, $S_g/t\cos\alpha = 3.0$, $F_{s,a} = 1.05$, $\alpha = 45°$ (for stator), $\alpha = 48°$ (for rotor), $St_{in} = 0.005$.

The specific work output of the turbine can be determined by mass and energy balance as below:

$$W_{gt} = m_{g,i}(h_{g,i} - h_{g,e}) + [\sum \dot{m}_{coolant}(h_{coolant,i} - h_{coolant,e})] - W_{ref} \quad (21)$$

PERFORMANCE PARAMETERS

The various parameters to evaluate the performance of GT cycle is as described below

The gas cycle specific work (W_{gc}) is given by

$$W_{gc,et} = W_{gt} - \frac{|W_c + W_{ref}|}{\eta_m} \quad (22a)$$

$$W_{plant} = \dot{m}.w_{plant} = \eta_{alt}.[W_{gt,net}] \quad (22b)$$

The gas turbine (topping cycle) efficiency (η_{plant}) is expressed as:

$$\eta_{plant} = \frac{W_{plant}}{\dot{m}_F.\Delta H_r} \quad (23)$$

work ratio which is defined as the ratio of turbine to compressor work is given as [21]

$$W_{rat} = \frac{W_{gt}}{|W_c|} \quad (24)$$

Heat Rate (HR) can be defined as the energy entering the system (kJ/h) divided by the work produced(kW).

$$\mathrm{HR} = \frac{3600.\mathrm{Q} \times \dot{\mathrm{m}}_{\mathrm{a}}}{\mathrm{W}_{\mathrm{plant}}} \quad (25)$$

A programme has been written in C++ language for allthe models and associated equations and results have been presented.

II. RESULTS AND DISCUSSION

Table.1 depicts the input parameters taken in our study. The gas-turbines are generally designed to function at ISO condition (T_a = 15º C and RH_a = 0.6). But here in this article the CIT and RH_a have been altered and ISO state is not possible to be achieved every time. Nevertheless the obtained results in this study have been validated with the features of MS 7001 GT from GE [22] at off design conditions and with IAC and is found to relate well.

In the present work the results of VCIAC have been studied for a cooled GT cycle by altering various factors and the results attained have been examined as under:

Effect of vapor compression inlet cooling and film bucket cooling on plant efficiency and plant specific work of BGT cycle.

Fig. 2 shows the effect of vapor compression inlet air cooling and air film bucket cooling on performance of gas turbine cycle at T_b=1123 K, $r_{p,c}$=23, T_a=313 K, RH_a=0.2. It has been observed that the integration of VCIAC increases the plant specific work and plant efficiency of BGT cycle. This is because for inlet air cooled cycle, the CIT is lower resulting in lower compression work. Also the mass flow rate of air increases due to increase in density. Both of these factors increase plant specific work. Though the fuel energy input increases at lower CIT, it is well compensated by increase in plant specific work and hence plant efficiency increases. It has been observed that the effect of vapor compression inlet air cooling is more pronounced in case of FCGT cycle as compared to BGT cycle. At a T_a = 313 K, RH_a = 0.2, $r_{p,c}$=23 and T_b= 1123 K, the integration of vapor compression inlet air cooling increases the plant efficiency of BGT cycle by 1.54 % and plant specific work by 8.61 %, against 2.21 % and 10.99 % respectively for FCGT cycle. This is because, in case of FCGT cycle, the addition of vapor compression inlet cooling has the additional benefit of lowering the bucket coolant requirement and associated losses.

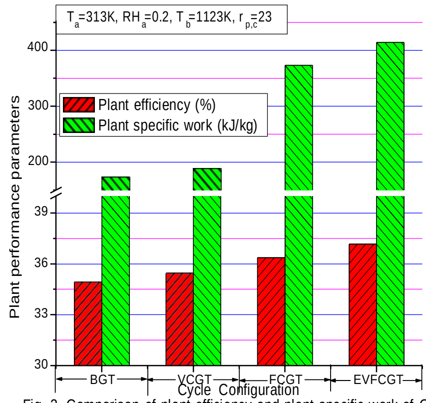

Fig. 2. Comparison of plant efficiency and plant specific work of GT cycle configurations

Variation of refrigeration work w.r.t. ambient temperature and ambient RH for VCFCGT configuration

Fig.3 (a) depicts the influence of ambient temperature on input refrigeration work for VCFCGT configuration. The divergence found in refrigeration work explains that ambient temperature has a desirable effect on refrigeration work output. This may be due to extraction of large amount of heat from inlet air resulting increase in refrigeration effect and work of refrigeration as well.

Fig.3 (b) illustrates the effect of ambient RH on refrigeration work for VCFCGT configuration. The histogram represents the gradual increase of refrigeration with increase in ambient relative humidity. This may be due to increase in cooling load.

Effect of CIT on mass of bucket coolant required and mass of fuel requirement for VCFCGT configuration

Fig.4 shows the potential influence of CIT on mass flow of bucket coolant as well as fuel required for VCFCGT configuration. It is observed that CIT is almost linearly decreases with decrease in mass flow rate of bucket coolant. As because, temperature drop of bled coolant causes decrease in CIT. However the drop in CIT becomes undesirable for CC to produce required amount of compressor power. So in order to maintain the temperature ,excess amount of heat need to be supplied to CC from external source.

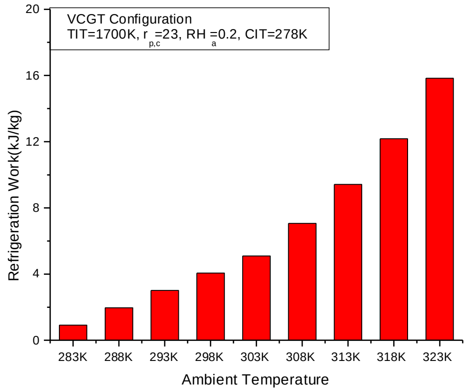

Fig. 5(a): Variation of refrigeration work w r to ambient temperature for VCGT configuration

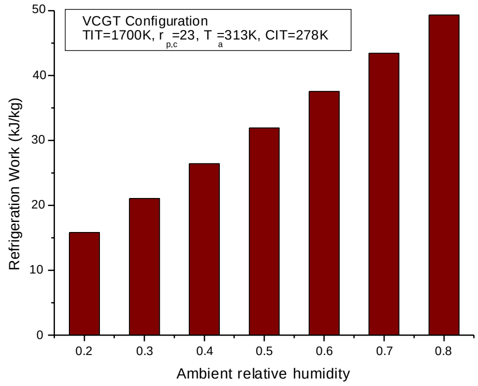

Fig. 5(b): Variation of refrigeration work w r to ambient RH for VCGT configuration

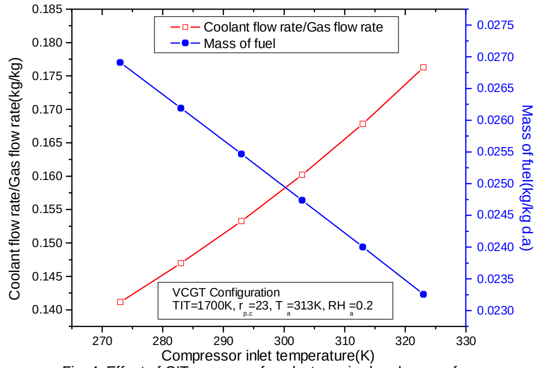

Fig. 4: Effect of CIT on mass of coolant required and mass of fuel requirement for VCGT configuration

Effect of ambient conditions on plant efficiency and plant specific work and of gas turbine configuration with and without VCIAC

The sensitivity histogram in Fig 5(a) ,explains the positive effect of ambient temperature on specific work output of gas turbine cycle engaged for with and without inlet air cooling. The exit temperature of inlet air cooling system is adhered to maintain a constant at 278K. It is observed that inlet air cooling gas turbine produces higher specific work output than that of cooled gas turbine cycle. This may be due to lower consumption of compressor work maximizes the net output of gas turbine.

The sensitivity histogram in Fig 5(b) shows the effect of ambient temperature on the gas turbine cycle efficiency on FCGT configuration with and without inlet cooling. It is observed that turbine gas cycle efficiency increases linearly with inlet air cooling. Gas cycle specific work output played a significant role in order to compensate the energy loss due to higher difference among CIT and PIT.

It is also observed that higher ambient temperature and lower relative humidity maximizes the plant efficiency as well as gross power output of an integrated inlet air cooling system. Moreover at higher ambient temperature, effective cooling can be achieved due to sufficient temperature difference between DBT and WBT. Similar trend of higher ambient and RH has been followed in order to enhance the refrigeration work as discussed in section4.3 reducing the gain of increase in plant specific work.

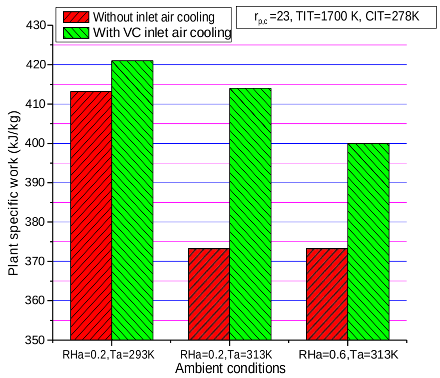

Fig. 5(a): Effect of ambient conditions on the plant specific work with and without VCIAC

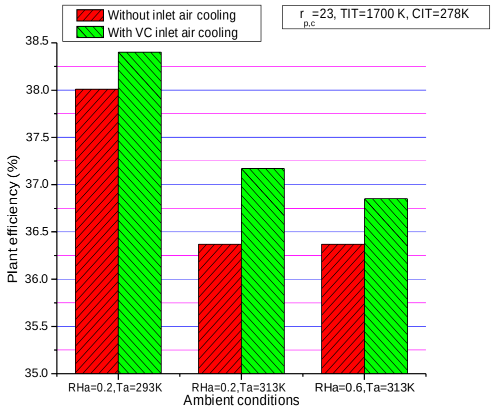

Fig. 5(b): Effect of ambient conditions on the plant efficiency with and without VCIAC

Effect of CIT on plant efficiency and plant specific work of VCFCGT configuration.

Fig 6 shows the effect of CIT on plant specific work and plant efficiency of VCFCGT configuration. It is observed that CIT decreases adversely with increase in plant specific work and plant efficiency of VCFCGT cycle. As discussed in section 4.3, a reduction in CIT leads to increase the plant specific work output. The reduction in Compressor inlet temperature results an increase in fuel energy input which may be due to temperature difference outlet temperature of compressor and inlet temperature of turbine. It is also observed that while a drop by 40K below the given ambient temperature of 323 K, the plant efficiency and specific work output increases by 4.46% and 14.59% respectively.

Performance map of plant efficiency and plant specific work of VCFCGT configuration for variation in $r_{p,c}$ and TIT.

Fig 7 shows the performance mapping of of gas turbine with design parameter such as r $_{p,c}$,TIT and cooling methods which becomes helpful for attaining the best plant efficiency and specific work. The result reflects the respective optimum TIT corresponds to maximum plant efficiency under all pressure ratios.In addition, compressor work input, fuel charging and bucket coolant air requirement increases with rise in r $_{p,c}$ and TIT. However, the gas turbine work decreases by increasing pumping ,cooling and mixing losses.

Effect of $r_{p,c}$ and r_{IT} on Heat rate and Work ratio of vapor compression inlet cooled gas turbine cycle

Fig 8 shows the effect of variation of r_{IT} and $r_{p,c}$ on heat rate and work ratio. It can be observed that the work ratio increases with increase in value of r_{IT}upto 5.6 after which it starts decreasing.

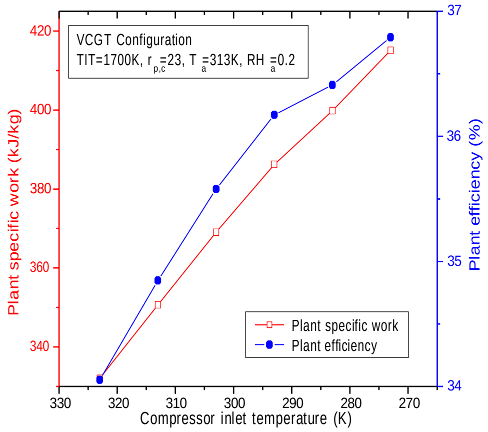

Fig. 6 Effect of CIT on plant specific work and plant efficiency of VCGT configuration

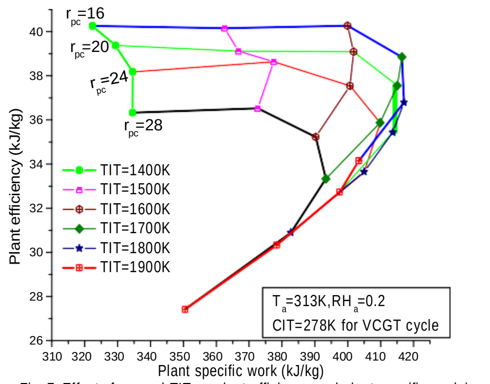

Fig. 7: Effect of $r_{p,c}$ and TIT on plant efficiency and plant specific work in a VCGT cycle.

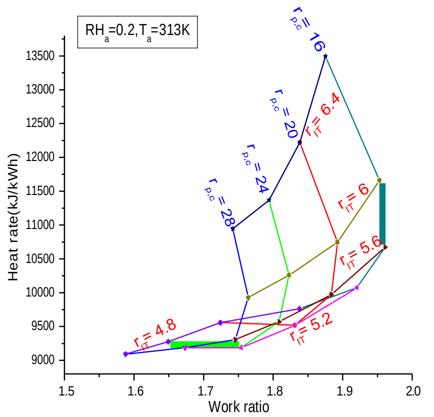

Figure 8. Effect of $r_{p,c}$ and r_{IT} on work ratio and heat rate in an VCGT cycle.

This is because a higher value of r_{IT} suggests an increase in TIT and/or a decrease in CIT both of which results in an increase in net gas turbine output. Beyond a $r_{IT}$of 5.6 however, the benefits of higher temperatures in terms of excess of gas turbine work over compressor work may be more than offset by the increased losses associated with the cooling flow rates required to cool the gas turbine buckets and so work ratio decreases. The work ratio and heat rate decreases with increase in $r_{p,c}$. There exists an optimum r_{IT} at any $r_{p,c}$ with reference to maximum work ratio. The decrease in heat rate with increase in $r_{p,c}$ is more pronounced at higher value of r_{IT} .The effect of variation of r_{IT} on heat rate suggests that the heat rate increases with increase in r_{IT} because of the increase in fuel energy input owing to higher difference between CIT and TIT that is unable to offset the gain in work output.

III. CONCLUSIONS

Based on the analysis of vapor compression inlet air cooled gas turbine cycle, the following conclusions have been drawn:

1. The addition of vapor compression inlet air cooling has been observed to improve the plant specific work and plant efficiency of gas turbine cycle.
2. Compression inlet air cooling increases the plant efficiency of BGT cycle by 1.54 % and plant specific work by 8.61 %, against 2.21 % and 10.99 % respectively for FCGT cycle.
3. It is also observed that higher ambient temperature and lower relative humidity maximizes the plant efficiency as well as gross power output of an integrated inlet air cooling system.
4. The increase in performance parameters due to integration of vapor compression inlet air cooling is superior in case of cooled gas turbine based combined cycle as compared to uncooled cycle.
5. The plant efficiency and specific work output increases by 4.46% and 14.59% respectively for a 40 K drop in ambient temperature.
6. The work ratio which represents the excess of turbine work over work of compression,has been observed to increase with increase in r_{IT} and decrease in $r_{p,c.}$
7. The work ratio increases with increase in value of r_{IT} upto 5.6 after which it starts decreasing. There is an optimum r_{IT} at any $r_{p,c}$at which the work ratio is maximum . The heat rate increases with increase in $r_{IT}$and decrease in $r_{p,c.}$.
8. For all values of TIT, there is an optimum $r_{p,c}$at which the plant efficiency is maximum.

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Nomenclature

c	Bucket chord
c_p	Specific heat capacity at constant pressure ($kJkg^{-1}K^{-1}$)
c_v	Specific heat capacity at constant volume ($kJkg^{-1}K^{-1}$)
f	Factor
f_{sa}	Correction factor to account for actual bucket surface area
gt	Gas-turbine
h	Specific Enthalpy (kJ kg^{-1})
ΔH_r	lower heat value, heat of reaction / combustion (kJ kg^{-1})
m	Mass flow rate (kJ kg^{-1})
p	Pressure (kJ kg^{-1})
q	Heat energy (kJ kg^{-1})
$r_{p,c}$	Compressor pressure ratio
S_b	Bucket perimeter at pitch line distance
st	Steam-turbine
Sn_{in}	Stanton number at stage inlet condition
T	Temperature (K)
W	power (kJ s^{-1})
w	Specific work (kJ kg^{-1})
η_{cycle}	Combined-cycle efficiency
X	Exergy [kJs^{-1}]
X_{dest}	Exergy destruction [kJs^{-1}]

SUBSCRIPTS

a	Air/ambient
alt	Alternator
b	Bucket
comp	Compressor
comb	Combustion
cc	Combustion Chamber
dest	Destroyed
ex	Exergy
e	Exit
f	Fuel
g	Gas
gt	Gas-turbine
i	Inlet
j	Bleed point
m	Mechanical
ph	Physical
pt	Polytropic
s	Steam
v	Volume (m^3)
w	Water

GREEK SYMBOLS

ε	Effectiveness
α	Gas flow discharge angle (degree)
γ	Specific heat ratio
ϕ	Relative humidity (ratio)
ω	Specific humidity (ratio)
η	Component efficiency (%)
ψ	S pecific exergy [$kJkg^{-1}$]
ξ_E	Exergy Destruction ratio (%)
ν	Specific volume ($m^3\ kg^{-1}$)
Δ	Delta or difference

ACRONYM

C	Compressor
CC	Combustion Chamber
BGT	Basic Gas Turbine
CIT	Compressor inlet temperature
GT	Gas-turbine

IAC	Compressor inlet cooling/Cooled
RH	Relative humidity
TIT	Turbine Inlet Temperature
VAIAC	Vapor absorption inlet air cooling/cooled
VAGT	Vapor absorption inlet air cooling/cooled gas turbine

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