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Air Craft Gas Turbines: A Thermodynamic Analysis

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Abstract
In the present paper the environmental impact of a gas-steam combined cycle, in terms of CO_2 emissions has been supplemented with the energetic analysis of the cycle. The gas turbine based triple-pressure reheat combined cycle incorporates, vapor compression inlet air cooling and air-film turbine blade cooling, to study the improvement in plant performance and sustainability. A parametric study of the effect of compressor pressure ratio ($r_{p,c}$), compressor inlet temperature (CIT), turbine inlet temperature (TIT), inlet temperature ratio (r_{IT}), ambient relative humidity and ambient temperature on performance and sustainability has been carried out. The integration of inlet air cooling and gas turbine blade cooling results in a significant reduction in CO_2 emission per unit plant output. The integration of vapor compression inlet air cooling to gas turbine based combined cycle, has been observed to improve the specific work by more than 10 %. The plant efficiency increases significantly with increase in TIT. For all values of TIT, there exists an optimum $r_{p,c}$ at which the plant efficiency is maximum. The cost of environmental impact due to CO_2 emission reduces with increase in TIT and decrease in CIT.

I. INTRODUCTION

Gas turbine-based combined cycle technology developed in the 1960s as a source to magnify gas turbine-based power plants efficiency has evolved considerably reaching today exceeding 60% without using any performance-enhancing methods such as power augmentation using inlet cooling, steam injection, or modifying gas turbine cycle such as interstage cooling, reheating and recuperation. This has been achieved through various means including but not limited to the advancements made in material technologies combined with high-temperature coatings allowing to achieve high turbine inlet temperature, improved blade cooling technologies, use of computational fluid dynamics methods to improve the aerodynamic performance of compressor and turbine sections, improvements in combustion technologies. However, the use of aforesaid approaches achieving performance enhancement has some consequences. Increased emissions levels is the first major effect of increasing turbine inlet temperature. Secondly, cooling air requirements for hot gas path components (such as combustor and turbine blades) increase impacting compressor section design. s, gas turbines have inherent design limitations as its performance reduced with the increment of ambient temperature. A given machine, depending on its design could lose as much as 0.5 to 0.8% of the power with 1 ^{0}C rise in ambient temperature. With the saturation in technological advancement of gas turbine, both gas turbine suppliers and users have to develop innovative approaches for improving plant performance and reducing GHG emissions and air pollutions. Current awareness of the environment and the concern for its protection has warranted the introduction of environmentally benign technologies. This creates an incentive to attempt for energy efficiency programs of natural gas-fired combined-cycle plant. Several procedures have been studied, analyzed and claimed to achieve an increase in the efficiency of gas turbines and combined cycles.

Considering the two seasons mainly summer and winter N. Asgria et al [1] have analyzed a trigeneration system in those two seasonal periods. The system was driven by a gas turbine integrated with an MSW gasification unit and a Li-Br Water absorption refrigeration cycle. They have found that the system performance is lower in winter and in summer the heating cycle contains more moisture in the produced gas. The fuel consumption is higher in summer in comparison to winter. Various cooling and heating heat exchangers present in the system results in a high amount of exergy destruction. The system can be enhanced by exergo-economic optimization in the future.

In search of alternate and clean options, Faustino Moreno-Gamboa et al [2] have worked on a single-stage hybrid Central Solar Power (CSP) plant based on Colombian weather conditions. They have used the DNI model to estimate the hourly solar radiation, Mean Squared Error and the Means Absolute Bias Error to get the result in acceptable values. They have implemented both energy and exergy analysis to estimate the CSP plant operation. The use or not of a regenerator and hybridization system was also considered while running the plant. Thus, the model can be made to work with various configurations.

System running with the combination of solid oxide fuel cell with gas turbine have been analyzed by MiladBeigzadeh et al [3] to reduce the pollution along with high-efficiency power output. Bio-fuels of various origins have been compared here. While comparing different operating parameters, energy and exergy efficiency, and the amount of irreversibility shown by various components of the system have also been analyzed. The comparison shows that the natural gas fed systems have higher efficiency with respect to the biogas fed system.

While M.F. Ezzat and I. Dincer [4] have developed a multigeneration energy system with ammonia fed solid oxide fuel cell and ammonia gas turbine to find an alternate environmental clean option of power. The first and second law of thermodynamics is being used as the principal law for assessment of the system. The overall energy and exergy efficiencies of the system were found to be 58.85% and 50.6% respectively. The useful outputs of the system were found to be electricity and hot water for residential application, cooling for industrial application, and hydrogen.

Taking offshore petroleum and gas platforms into consideration F.C.N. Silva et al [5] have compared an S-Graz oxyfuel cycle with a conventional open gas turbine cycle plant. The platform performance is being analyzed by the rate of exergy destruction. The environmental impact is being studied by computing the amount of CO_2 emitted from the platform. The energy integration analysis revealed heat recovery opportunities, exergy cost, and specific CO_2 emitted. While an air separation unit, a chemical absorption unit and a compression battery for CO_2 re-injection purposes seem to be more competitive, the conventional configuration is efficient. Coupling Oxyfuel with an extensive heat exchange network for energy integration can attenuate the system to reduce the emission of harmful gases. These configurations could be considered economical in many ways.

For over a period of one-year G. Caposciutti et al [6] have studied the performance of a mini gas turbine operating with biogas. The inlet temperature was reduced by introducing an absorption chiller. By the model developed in AMESim and considering the working period of one-year the study was done. Aspen Exchanger Design and Rating was used to develop all the heat exchanger. The simulation was done in AMESim by adopting an ε-NTU approach. The energy and economic analysis both were considered while assessing the performance. The economic profitability is highly influenced by the chiller size and geographical location based on temperature with medium-high constant temperature over the whole year. The study shows that the expected payback period may range between 9 or more years.

As proposed by Mohammad Reza MajdiYazdia et al [7] the power output of a gas turbine is greatly influenced by the 4E (energy, exergy, environmental and economic). Out of all the conditions, the environmental factor contributes more to the output power. Warm and dry climates favors the performance of absorption chiller system and the inlet fogging system with respect to humid climate. For locations where the temperature is more than the dew point temperature, the heat pump shows better performance. The energy efficiency of the system is higher and the fuel consumption is greatly reduced by adding the inlet fogging and absorption chiller system. These two systems also reduce the emission of NOx and CO2 gases.

By introducing a Variable Inlet Guide Vane (VIGV) schedule along with the inlet air cooling the thermal efficiency and specific fuel consumption can be improvised. This was investigated and analyzed by Muhammad BaqirHashmi et al [8]. The cumulation of both inlet air cooling and VIGV showed promising result which was earlier deteriorated by the surge of ambient air temperature and relative humidity. With these suggestions, the industrial gas turbine can be upgraded for enhanced performance and durability even in dry and humid climates.

Humidity and ice blockages in winter climate have created an opportunity for Shucheng Wang et al [9] to modify a Combined Cycle Power Plant by adding inlet air heating to it. They have modeled the system using Ebsilon software and validated its experimental results. The use of heat recovery steam generators has raised the overall plant efficiency. And the exergy destruction is also being reduced by the inlet air cooling. In both Brayton and Rankine cycle, the exergy destruction was also reduced by inducing the heated inlet air.

ShahrulNahar Omar Kamal et al [10] have studied the performance of gas turbines in the Malaysian climate where they found that the TIAC technology used for power augmentation for the Gas Turbine proved to be reliable. They have investigated an LM6000PD Gas Turbine Generator. The net power output is found to be increased within a range of 27.5% to 32.11% and the net heat rate is being reduced within a range of 2.8% to 3.74%. The validation was being done by using GT Pro software. The results showed that the Turbine Inlet Cooling using an Electric chiller as a power augmentation technology is effective in Malaysia.

Based on the study conducted by Anoop Kumar Shukla et al [11], on a Gas Turbine employed with vapor compression inlet air cooling the output is a function of ambient air temperature and ambient relative humidity. With an increase in ambient air temperature, the thermal efficiency and specific work output decreased. The difference of power they have found between a gas turbine with vapor compression and a gas turbine without it is 20.769 kJ/kg.

Hyun Min Kwon et al [12] have studied an H-class gas turbine. They proposed to cool the hot parts of the gas turbine along with inlet air heating. They found the maximum degree of coolant pre-cooling and inlet air cooling using a commercial absorption chiller was 45 K and 11 K respectively. A power boost of 8.2 % was also obtained with the implementation of dual cooling. The economic feasibility was also observed found to be around 7.7 % of gross net profit.

With an unusual approach of using liquefied natural gas's cold energy to decrease the temperature of inlet air, Zuming Liu et al [13] have investigated its effect on a gas turbine. The system consists of two organic Rankine cycle in series with a mechanical vapor compression chiller. The results found in their investigation shows the larger air temperature drop of the range 2.1 – 5.3 oC and higher cooling duty of range 6.9 – 7.3 MW. The cooling system improves the overall power plant output by 0.79 – 3.04% that is around 2.8 – 11.1 MW. The economic analysis shows that the NPV was $1.6 – 34.4 million higher than the predicted value.

A novel method applied by Yongping Yang et al [14] where the gas turbine is being retrofitted with a compressor inlet air heating process for part-load performance improvement. The inlet air was preheated by the flue gas from the heat recovery steam generator. By the application of preheated air, the performance increment was found to be 1.7% pt. They observed the specific work decreased due to the specific work being consumed by the compressor. The overall cycle performance and the fuel consumption rate is improvised by increasing the temperature of the compressor outlet air which in turn reduces the extra exergy destruction during the combustion process. This can be used as an innovative solution for gas turbine combined cycle load management.

With the combination of Gas Turbines along with desalination unit, thermal vapor compressor, organic Rankine cycle with steam ejector refrigeration, and heat exchangers H. You et al [15] have conducted a performance assessment of gas turbines. They have developed a mathematical model and done the thermodynamic energy exergy analysis of the system. The parametric study is also being conducted under various load conditions. The power output can be increased by precooling the inlet air of the air compressor and utilizing the energy cascade system. The constraint to be overcome as suggested by them was the exergy destruction occurring in the combustion chamber.

V. Zare [16] have applied inlet air cooling to a conventional open cycle gas turbine power plant. The waste heat produced from the exhaust gas is being used to run an ammonium-based absorption refrigeration cycle for compressor inlet cooling. Thermoeconomic analysis of the system with and without inlet air cooling is being done. The result of implementing the let air cooling shows the improvement of 30.1%, meanwhile, the Levelized Cost of electricity was reduced by 22.5 %.

The Thermo economic analysis of a gas turbine cycle with supercritical carbon dioxide and organic Rankine cycle is being done by KasraMohammadi et al [17]. They have taken a triple power cycle. The waste heat from the gas turbine is reused in running a supercritical carbon dioxide recompression cycle and a recuperative organic Rankine cycle in sequence. They have used MATLAB to evaluate the performance and optimized the system by particle swarm optimization algorithm. The most influential parameters for the thermoeconomic performance analysis were the inlet temperature of the gas turbine and the supercritical carbon dioxide cycle.

Micro gas turbines have been extensively studied by Gustavo Bonolo de Campos et al [18]. Although the efficiency is relatively low, the performance can be enhanced by coupling it with an organic Rankine cycle. Saturation temperature at the ambient pressure is an important variable. By constraining the minimum pressure above ambient and a high degree of superheating is being favored the recuperated cycle is heated directly under microturbine flue gases. Pentane is found to be the most suitable working fluid with high power and overall efficiency in comparison to others. The author found 35% as the upper practical thermal efficiency for their system.

Thermophysical and thermochemical recuperation based on the methane steam reforming process is being investigated by M. Sadeghi et al [19]. The performance of the combined gas cycle turbine and organic Rankine cycle is being analyzed. Along with the thermo-economic analysis, multi-objective optimization using a genetic algorithm is being conducted for the system and finally the system characteristics are being compared. As the combustion temperature raises the thermodynamic, economic, and environmental performances are also enhanced. The performance is ranged between 22.86% - 27.52% based on exergy analysis. However, the production cost was found to be lower than other configurations.

While comparing Inter-cooled gas turbines and basic gas turbine cycle thermoeconomicallyMithilesh Kumar Sahu and Sanjay [20] have found that the intercooled gas turbine requires 47.34% higher fuel during operation and delivers about 40 % higher specific work. They have developed a computer code using various models comprising of governing equations. The code was used to calculate power utility and operating parameters. The analytical model used contains a blade cooling phenomenon. For the current investigation, they have adopted the average cost theory approach out of many thermoeconomic methodologies.

Despite numerous research on inlet air cooling the literature review reveals the following.

- All the studies made so far represent the effect of vapor compression inlet-cooling on the combined cycle without considering the effect of blade coolant mass on its performance.
- None of the previous literature has reported the effect of vapor compression inlet-cooling on triple pressure HRSG based combined cycle.
- The study of the effect of ambient temperature, ambient relative humidity, compressor pressure ratio, turbine inlet temperature, and inlet temperature ratio on the performance of vapor compression inlet-cooling systems integrated to cooled gas turbine-based combined cycle plant has not been investigated in literature.
- The environmental impact of a gas-steam combined cycle subjected to inlet air cooling and turbine blade cooling has not been discussed in the literature.

The present article overcomes the above-detailed research gap by analyzing the effect of integration compression inlet air cooling schemes to an air film turbine bucket cooled combined cycle plant. The emission analysis and the environmental impact have been discussed and the cost associated with environmental damage is presented for the combined cycle with and without inlet air cooling.

II. MODELING OF GOVERNING EQUATIONS

Parametric study of the combined cycle using different means of cooling has been carried out by modeling the various elements of a combined cycle and using the governing equations. The following are the modeling details of various elements. Figure 1 shows the schematic diagram of a triple pressure HRSG combined cycle with vapor compression inlet cooling system.

2.1 Gas Model

The specific heat of real gas varies with temperature and also with pressure at extreme high pressure levels. However, in the present model it is assumed that specific heat of gas varies with temperature only and is given in the form of polynomials as follows

$$c_p(T) = f_h * \left[a + bT + cT^2 + dT^3 + \right] \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \tag{1}$$

where a, b, c, and d are coefficient of polynomials, as taken from the work of Touloukian and Tadash (1970) and f_h is the humidity correction factor to account for the increase in specific humidity of ambient air across the air-humidifier

$$f_h = 1 + 0.05\phi_{h,e} \tag{2}$$

where $\phi_{h,e}$ is the relative humidity at the outlet of humidifier.

Thus, the enthalpy of gas is expressed as

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

2.2 Modelling of Vapor Compression Inlet Air cooling system

Fig. 2(a) illustrates the path that air takes in the psychometric chart as it undergoes change from ambient state (a) to the desired cooled state (c). By rejecting its sensible heat, the air temperature drops, while the relative humidity continues to rise until its dew point is reached (point ‘b’). Any further cooling from this point will require removal of a much greater quantity of heat due to the need of removal of latent heat of condensation of the water vapor along with the sensible heat that the air contains. This process continues until the desired temperature at point ‘c’ is achieved. The air cooling process has been plotted in Fig 2 with heavy lines a-b-c and the sensible and latent cooling loads are shown as d-c and a-d respectively.

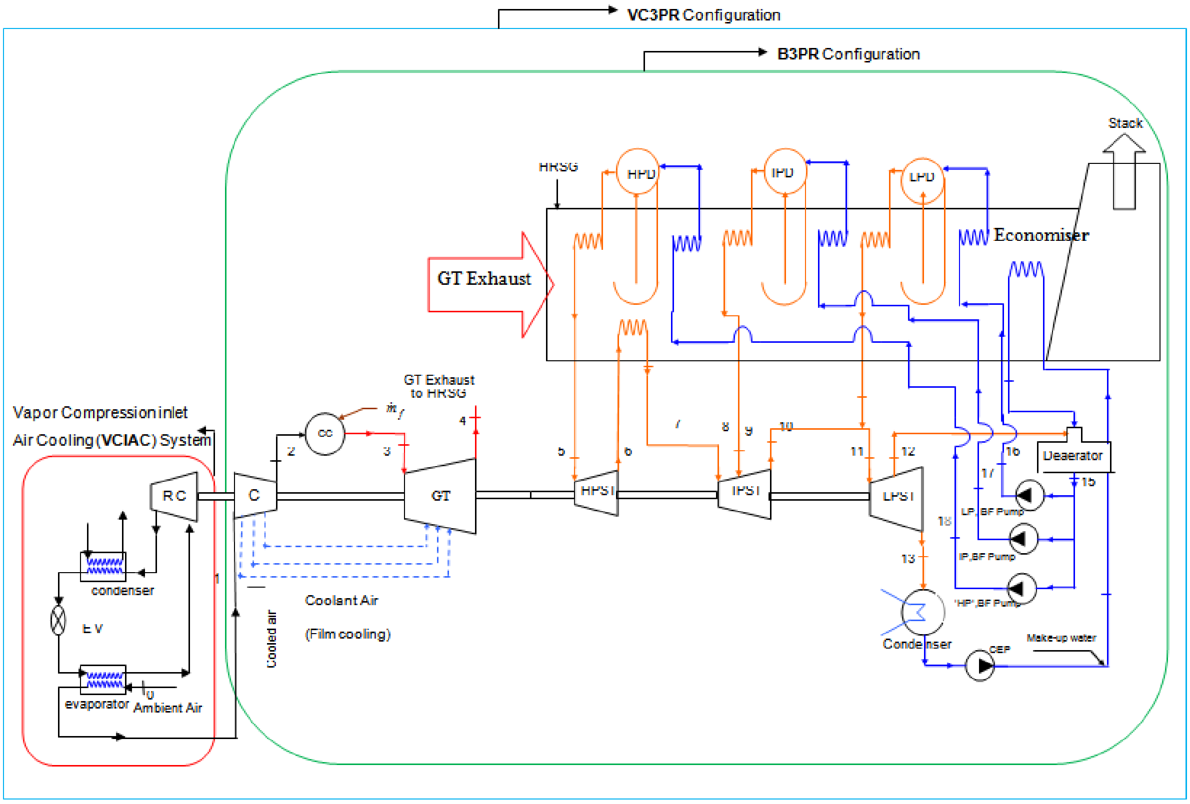
The total cooling load comprises of the heat required to reduce the temperature from its initial ambient condition to the desired cooled state, i.e. the sensible heat of air and the heat required to be removed, to condense the moisture contained in the air or the latent heat. So the total cooling load or the refrigerating effect can be computed as under:

$$Q_{CL} = \dot{m}_a \left(q_{sensible} + q_{latent}\right) \tag{3}$$

$$q_{CL,sensible} = h_d - h_c \tag{4}$$

$$q_{CL,latent} = h_a - h_d \tag{5}$$

h_a, h_c and h_d are the enthalpy of air at points a, c and d respectively.



A vapor compression refrigeration cycle has been used to cool the compressor inlet air. The work of refrigeration needed for cooling the inlet air has been extracted from the gas turbine output. The actual power required to run the refrigeration system is more than the theoretical power due to various associated losses. So the actual refrigerating compressor power is given by

$$W_{comp,ref} = \frac{\dot{m}_a (h_a - h_c)}{\eta_m \eta_{el} \eta_{vol}} = \frac{\dot{m}_a (h_a - h_c)}{\eta_{eu}} \tag{3}$$

Where η_m, η_{el} and η_{vol} are the compressor mechanical, electrical and volumetric efficiency respectively.

The refrigeration work for a selected COP of vapor compression refrigeration is calculated by using the following relation [27]

$$W_{ref} = \frac{Q_{CL}}{COP_{VC}(1-\mu\beta)^n \eta_{eu}} \tag{4}$$

where$_{\mu}$ is an empirical constant that depends on the type of refrigerant and β is the quality at the exit of IAC system. Freon 22 is choosen as the working fluid in the refrigeration machine for which the value of α is 0.77. The empirical constant n depends on the number of compression and expansion stages. In this paper the value of n=1 for a simple refrigeration cycle with single stage compressor. The compressor energy use efficiency η_{eu} is normally determined by manufacturers and depends on the pressure ratio, the application and type of the compressor.

In proposed cycle a vapor compression refrigeration cycle is used for cooling the compressor inlet air. The work of compression needed for cooling the inlet air is derived from the turbine output. The refrigeration work is optimised and represented in terms of t_o, t_1, ε_e and ε_c by visualising the actual refrigeration cycle as a realistic conceptual cycle in which heat transfer rates are constrained to be finite in both evaporator and condenser.

For this optimised refrigeration cycle work of refrigeration can be given by [2]

$$W_{comp,ref} = \frac{c_{p,a}(T_a - T_{c,i})\left[\varepsilon_e T_h - \left\{T_a \varepsilon_e - (T_a - T_{c,i})\right\}\right]}{\left[\eta_{ref} . \left\{T_a \varepsilon_e - (T_a - T_{c,i})\right\}\right]}$$

Where $T_h = \frac{\left[\left(3T_a^2 + T_{c,i}^2\right) + \sqrt{\left\{\left(3T_a^2 + T_{c,i}^2\right)^2 - 4T_{c,i}T_a\left(T_a + T_{c,i}\right)\left(3T_a - T_{c,i}\right)\right\}}\right]}{2\left(T_a + T_{c,i}\right)}$

2.3Compressor Model

The compressor used in gas turbine power plant is of axial flow type. The thermodynamic losses in an axial flow compressor are incorporated in the model by introducing the concept of polytropic efficiency.
Reference [17] gives detail discussion on modeling and governing equations.
Using mass and energy balance across control volume of compressor, the compressor work is calculated as follows:

$$\dot{m}_{c,i} = \dot{m}_{c.e} + \sum \dot{m}_{coolant,j} \tag{5}$$

$$W_c = \dot{m}_{c,e} h_{c,e} + \sum \dot{m}_{coolant,j} h_{coolant,j} - \dot{m}_{c,i} h_{c,i} \tag{6}$$

2.4 Combustor Model:

Losses inside the combustor, which arise due to incomplete combustion and pressure losses, are taken into account by introducing the concept of combustion efficiency and percentage pressure drop of compressor exit pressure [Table 1]. The mass and energy balances across the control volume of combustor yield the mass of fuel required to attain a specified exit temperature of combustor which is taken as turbine inlet temperature (TIT), given by,

$$\dot{m}_e = \dot{m}_i + \dot{m}_f \tag{7}$$

$$\dot{m}_f \cdot \Delta H_r \cdot \eta_{comb} = \dot{m}_e \cdot h_e - \dot{m}_i \cdot h_i \tag{8}$$

The combustion reaction and the combustion products of hydrocarbon fuel ($C_{a1}H_{b1}$) are expressed as

$$\begin{aligned} &C_{x_1}H_{y_1} + \xi \cdot \left(x_{o_2}O_2 + x_{N_2}N_2 + x_{H_2O}H_2O + x_{CO_2}CO_2\right) \\ &\rightarrow \xi \cdot \left(y_{CO_2}CO_2 + y_{N_2}N_2 + y_{o_2}O_2 + y_{H_2O}H_2O\right) \end{aligned} \tag{9}$$

Where

$$y_{CO_2} = \frac{x_1}{\xi} + x_{CO_2} \tag{10}$$

$$y_{N_2} = x_{N_2} \tag{11}$$

$$y_{H_2O} = \frac{y_1}{2\xi} + x_{H_2O} \tag{12}$$

$$y_{O_2} = x_{CO_2} - \frac{x_1}{\xi} - \frac{y_1}{4\xi} \tag{13}$$

$$\xi = \frac{n_a}{n_f} \tag{14}$$

2.5 Cooled Gas Turbine

In this work, the gas turbine blades have been modeled to be cooled by air-film cooling (AFC) method. The cooling model used for cooled turbine is the refined version of that by Louis et al [9]. he detailed modeling is described in author’s earlier article [20]
The Turbine work is given by the mass and energy balance of gas turbine as under:

$$W_{gt} = \left[\dot{m}_{g,i} \cdot (h_{g,i} - h_{g,e})\right] + \left[\sum \dot{m}_{coolant} \cdot (h_{coolant,i} - h_{coolant,e})\right] - W_{ref} \tag{15}$$

2.6 HRSG Model

In the present study, the bottoming cycle employs an unfired triple pressure HRSG with reheat (3PR). The selection of steam pressure and temperature for 3PR system is based on the optimized value to yield maximum steam cycle efficiency satisfying the minimum stack temperature and quality of steam at ‘lp’ turbine exhaust. Reference [18] outlines the detailed procedure.

2.7 Steam turbine model

In the present work, there are three main steam turbine stages, namely HPST, IPST and LPST. The inefficiency of steam turbine is incorporated in the model by introducing the concept of pressure loss due to throttling and isentropic efficiency (Table 1). Mass and energy balance yield steam turbine output given by Eq. below.

$$W_{st} = \sum^{stages} \dot{m}_{s.i} \cdot \left(h_{s,i} - h_{s,e}\right)$$

(16)

$$h_{s,i} - h_{s,e} = \eta_{st}\left(h_{s,i} - h_{s,e}\right)_{isentropic}$$

(17)

where $\dot{m}_{s.i}$ = amount of steam entering to the respective main turbine stages as per configuration.

2.8 Condenser model

The inefficiency in condenser due to pressure and heat losses causes undercooling of the condensate. This loss is accounted by an appropriate assumption of the value of undercooling that takes care of the losses (Table 1). The mass and energy balance give the cooling requirement as below.

$$\dot{m}_{w,i} = \dot{m}_{w,e} \text{ and } \dot{m}_{s,i} = \dot{m}_{cond,e} \tag{18}$$

$$(\dot{m}_{s,i}.h_{s,i} - \dot{m}_{cond,e}.h_{cond,e}) = \dot{m}_{w}.(h_{w,e} - h_{w,i}) \tag{19}$$

2.9 De-aerator model

Deaeration of feedwater in deaerator removes its dissolved oxygen content. Deaerator, also acts as a direct contact feedwater heater. It is assumed that there is negligible undercooling of the exit feedwater. The mass and energy balance give the amount of steam extracted for deaeration such as,

$$\dot{m}_{s,d/a,i} + \left(\dot{m}_{cond} - \dot{m}_{s,d/a,i}\right) = \dot{m}_{s,d/d,e} \tag{20}$$

$$\dot{m}_{s,d/a,i}.h_{s,d/a,i} + \left(\dot{m}_{cond} - \dot{m}_{s,d/a,i}\right)h_{s,cond,e} = \dot{m}_{s,d/d,e}.h_{s,cond,e} \tag{21}$$

It needs to be mentioned that in deaerator $\dot{m}_s$ is in liquid form.

2.10 Pump model

The mass and energy balance yields the pump work input needed:

$$\dot{m}_{w,i} = \dot{m}_{w,e} \tag{22}$$

$$W_p = \sum v_{w,i}.\left(p_e - p_i\right) \tag{23}$$

III. PERFORMANCE PARAMETERS:

The general expressions to calculate the performance parameters for gas turbine and combined cycle plant are given as follows:

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

$$W_{gc,net} = W_{gt} - \frac{\left|W_c + W_{ref}\right|}{\eta_m} \tag{24}$$

The steam cycle power (W_{sc}) is given by:

$$W_{sc,net} = W_{st}.\eta_m - \frac{\left|W_p\right|}{\eta_p} \tag{25}$$

where ‘η_p’ is the overall pump efficiency

The combined cycle power (W_{plant}) is given by:

$$W_{plant} = \eta_{alt} \cdot \lfloor W_{gc,net} + W_{sc,net} \rfloor \tag{26}$$

The combined cycle plant efficiency (η_{plant}) is expressed as:

$$\eta_{plant} = \frac{W_{plant}}{\dot{m}_f \cdot \Delta H_r} \tag{27}$$

Another parameter of great importance to the gas turbine is the *work ratio* [28]

$$W_{rat} = \frac{W_t}{|W_c|} \tag{28}$$

This parameter should be as large as possible, because a large amount of the power by the turbine is required to drive the compressor, and because the engine net work depends on the excess of the turbine work over the compressor work.

Heat Rate (HR) is the rate of heat input (kJ/h) required to produce unit power output (1kW).

$$HR = \frac{3600 Q \times m_a}{W_{plant}} = \frac{3600}{\eta_{plant}} \frac{kJ}{kWh} \tag{29}$$

The CO_2 emission per unit plant output is given as:

$$\kappa = \frac{\dot{m}_{CO_2} \times HR}{LHV} \frac{kg}{kWh} \tag{30}$$

Where $\dot{m}_{CO_2}$ = Mass of CO_2 produced in kg per kg fuel.

Cost rate of environmental impact due to CO_2 emission

$$C_{env,CO_2} = \dot{m}_{CO_2} \times \dot{m}_f \times C_{CO_2} \ \$/\mathrm{h} \tag{31}$$

Where $\dot{m}_f$ = Mass of fuel supplied in kg per hour.

Modeling of cycle components and governing equations developed for cycles proposed above have been coded using C++ and results obtained. The input data used in the analysis is given in Table 1.

IV. RESULTS AND DISCUSSION

In the present study the environmental impact in terms of CO_2 emissions is integrated with the energetic analysis of a evaporative inlet air cooled 3PR combined cycle through the help of performance curves, plotted using modeling, governing equations and input parameters (Table1). The environmental impact is expressed here as the total cost rate of pollution damage ($/h) due to CO_2 emission (C_{env,CO_2}) by multiplying its flow rates by corresponding unit damage cost, which have been reported elsewhere as, C_{CO_2} = 0.024$/kg [30]. In the present work the cost of pollution damage is assumed to be added directly to other system costs.

It is assumed that the volumetric composition of the inlet air is 0.7567 N_2, 0.2035 O_2, 0.003 CO_2 and 0.036 H_2O [30]. To improve environmental sustainability, it is necessary not only to use sustainable or renewable sources of energy, but also to utilize non-renewable sources like natural gas fuel more efficiently, while minimizing environmental damage. In this way, society can reduce its use of limited resources and extend their lifetime. CO_2 accounts for about 50% of the anthropogenic greenhouse effect. A major energy efficiency program would provide an important means of minimizing greenhouse effect through reduced CO_2 emission.

4.1 Effect of ambient temperature and ambient RH on the difference between DBT & WBT.

Fig 2 shows the effect of ambient temperature and ambient relative humidity on the effectiveness of evaporative inlet air cooling in terms of excess of DBT over WBT. It can be observed that the difference between dry and wet bulb temperature increases with increase in ambient temperature as well as ambient relative humidity. This behavior can be attributed to the psychometric property of air [12]

4.1 Effect of ambient conditions on plant efficiency and plant-specific work and of 3PR configuration with and without VCIAC

Fig. 2 shows the effect of ambient conditions on the plant-specific work of 3PR configuration with and without VCIAC. It is observed that the plant-specific work of the VC3PR cycle is higher than for the B3PR cycle for the entire range of selected ambient conditions. This is because, at these values of ambient temperature and ambient RH, the gain is due to an increase in gas cycle work (due to lower CIT resulting in reduced work of compression and lesser blade coolant requirement) and decrease in gas cycle work (due to high-grade energy extracted from gas turbine to run the IAC system) is always positive.

This net gain is higher at higher ambient temperature and lowers at higher ambient RH. As at higher ambient temperature and lower at higher ambient RH, the difference between DBT and WBT is also higher resulting in more effective cooling.

Fig. 3 shows the effect of ambient conditions on the plant efficiency of 3PR configuration with and without VCIAC. It is observed that plant efficiency increases as a result of VC inlet air cooling. This is because the increase in plant-specific work as explained above for Fig.6 (a) is able to offset the effect of an increase of input fuel energy of VC3PR configuration due to higher difference between CIT and TIT except for higher ambient RH.

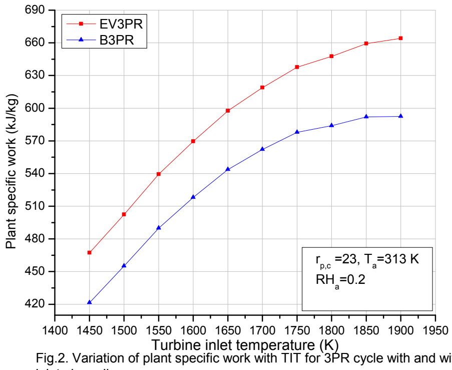


Fig.2. Variation of plant specific work with TIT for 3PR cycle with and without inlet air cooling

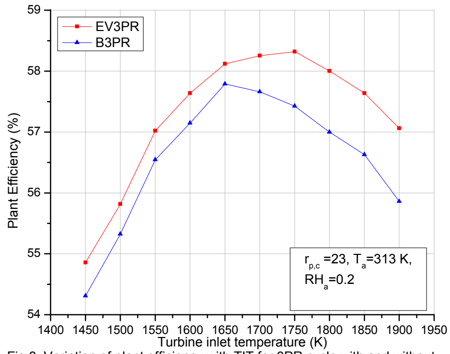


Fig.3. Variation of plant efficiency with TIT for 3PR cycle with and without inlet air cooling

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

Fig 4 shows the variation of compressor pressure ratio and the ratio of TIT to CIT on heat rate and work ratio. It can be observed that the work ratio increases with an increase in r_{IT} and a decrease in $r_{p,\,c}$. The effect of variation of r_{IT} on heat rate suggests that the heat rate first decreases up to a certain value of $r_{p,c}$ after which it starts increasing. There exists an optimum r_{IT} at any $r_{p,c}$ with reference to minimum heat rate. This optimum value of r_{IT} is found out to be 5.6 at an $r_{p,c}$ of 20 and 6 at an $r_{p,c}$ of 28K. The variation of heat rate with $r_{p,c}$ shows that at an r_{IT} of 6.4, the heat rate decreases with an increase in compressor pressure ratio. As the value of r_{IT} decreases this reduction in heat rate w.r.t $r_{p,c}$ gradually becomes less significant and beyond an r_{IT} of 5.2 this trend is reversed and the heat rate increases with an increase in $r_{p,c.}$ The optimum value of $r_{p,c}$ is found out to be 28 at an r_{IT} of 6.4 while it is 16 at an r_{IT} of 4.8. The existence of this optimum r_{IT} and $r_{p,c}$ is due to the combined effect of various factors. With an increase in both $r_{p,c,}$ and r_{IT}, the compressor work input, the fuel, and blade coolant requirement increases. The gas cycle work also increases but is restricted by increasing pumping, blade cooling, and mixing losses. The gas turbine exhaust temperature increases with an increase in r_{IT} but decreases with an increase in $r_{p,c}$ which decides the generation of steam in the HRSG and hence the steam turbine output.

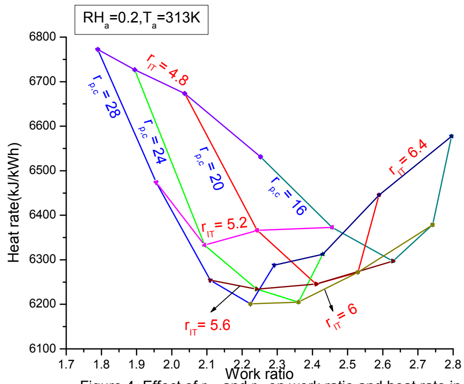


Figure 4. Effect of $r_{p,c}$ and r_{IT} on work ratio and heat rate in an VC3PR cycle.

The work ratio increases by 28.04% and heat rate reduce by 7.16% when r_{IT} increases from 4.8 to 6.0 at an $r_{p,c}$ of 28. Similarly, the heat rate reduces by 4.11% when the $r_{p,c}$ increases from 16 to 28 at an r_{IT} of 6.4.

4.3 Variation of CO_2 emission with r_{IT} and $r_{p,c}$ for VC3PR combined cycle.

Fig. 5 shows the variation of CO_2 emission per unit plant output of VC3PR combined cycle w.r.to r_{IT} and $r_{p,c.}$ A large reduction in CO_2 emission per unit plant output is observed with an increase in r_{IT} up to 6. This is because an increase in $r_{IT}$is obtained as a result of an increase in TIT and a decrease in CIT, both of which increase plant efficiency and help in better utilization of fuel. Beyond a r_{IT} of 6 however, the reduction in coolant flow due to lower CIT is unable to offset the rise in coolant flow requirement due to higher TIT resulting in a rise in heat rate and a corresponding increase in CO_2 emission per unit plant output.

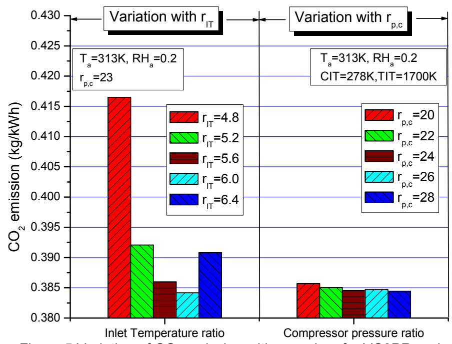


Figure 5.Variation of CO_2 emission with r_{IT} and $r_{p,c}$ for VC3PR cycle

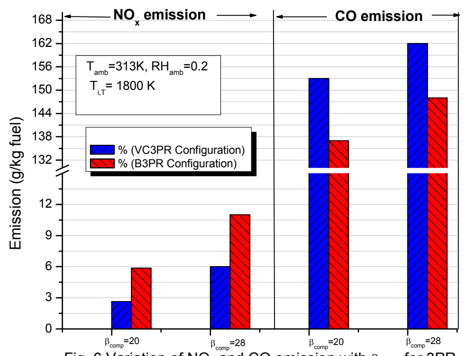


Fig. 6.Variation of NO_x and CO emission with β_{comp} for 3PR combined cycle with and without compressor inlet cooling.

The variation of CO_2 emissionper unit plant output with $r_{p,c}$ however is not appreciable rather monotonous. This is because an increase in $r_{p,c}$ has no significant effect on heat rate reduction.

4.4 Variation of NOx and CO emission with β_{comp} for 3PR combined-cycle with and without compressor inlet cooling.

Fig. 6 shows the variation with β_{comp} of NO_x and CO emission *of* 3PR combined-cycle for various means of compressor inlet cooling. It has been observed that the compressor inlet cooled cycle has less NO_xemission than the cycle without compressor inlet cooling, which signifies the importance of incorporation of these compressor inlet cooling schemes in terms of enhanced plant performance and hence reduced environmental damage. The NO_x emission is least for refrigerative inlet air cooling schemes because of lower combustor inlet temperature for a given value of β_{comp} As β_{comp} increases, there is a corresponding increase in combustor inlet temperature and hence adiabatic flame temperature, resulting in an increase in emission. The variation of CO emission with β_{comp} suggests that the COemission reduces with an increase in β_{comp} for all configuration. Its also observed that the CO emission is higher for inlet air cooling configuration but the combined effect of both the emissions is such that the total cost rate of pollution damage is lower for IAC configuration as discussed in the preceding section.

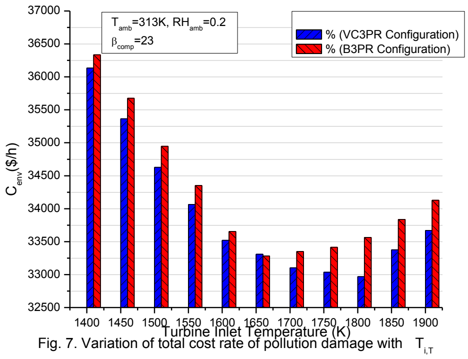


Fig. 7. Variation of total cost rate of pollution damage with $T_{i,T}$ of 3PR cycle with and without compressor inlet cooling.

4.5*Variation of cost rate of environmental impact due to CO_2 emission with $T_{i,T}$ for different means of compressor inlet cooling.*

Fig. 7 shows the effect of $T_{i,T}$ on environmental impact due to CO, NO_x, and CO_2 emission for constant net work output in terms of the total cost of pollution damage ($/h) by multiplying their respective flow rate to its corresponding unit damage cost, (0.02086 $/kg for CO, 0.024 $/kg for CO_2 and 6.853 $/kg for NO_x) [32]. The mass flow rate of air is assumed as 542kg/s [23]. It has been observed that the total cost of pollution damage reduces significantly with an increase in $T_{i,T}$ in spite of an increase in fuel consumption. This is primarily because with an increase in $T_{i,T}$the work output also increases and so fuel consumption reduces for constant work output which is considered in this case for analysis.

This optimum $T_{i,T}$ with reference to minimum cost rate of pollution damage is the same as that corresponding to the maximum efficiency. A higher value of $T_{i,T}$ can be adopted by adopting superior bucket cooling techniques. It can therefore be concluded that the collective effect of compressor inlet cooling and gas-turbine bucket cooling has a significant effect on sustainable development in terms of reduced greenhouse gas emission.

V. CONCLUSIONS

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

1. The plant-specific work of the VC3PR cycle is higher than for the B3PR cycle for the entire range of selected ambient conditions

2. This net gain is higher at higher ambient temperature and lowers at higher ambient RH

3. The Plant efficiency also increases as a result of VC inlet air cooling. This is because the increase in plant-specific work

4. The work ratio increases by 28.04% and heat rate reduce by 7.16% when r_{IT} increases from 4.8 to 6.0 at an $r_{p,c}$ of 28

5. The heat rate reduces by 4.11% when the $r_{p,c}$ increases from 16 to 28 at an r_{IT} of 6.4.

6. A reduction in CO_2 emission per unit plant output is observed with an increase in $r_{IT.}$

7. The NO_x emission reduces due to the integration of inlet air cooling and increases with an increase in pressure ratio.

8. The CO emission increases due to inlet air cooling but the total cost rate of pollution damage due to CO, $CO_{2,}$ and NO_xemission combined is lower for inlet air-cooled configurations.

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