# Performance and Exergetic analysis of a simple cycle gas turbine based power plant retrofitted with Inlet air Fog cooling (FCS) and Steam injection gas turbine (STIG)

Shyam Agarwal, B.B.Arora, Akhilesh Arora*, Rajat Pardal**
Department of Mechanical Engineering, Delhi Technological University, Delhi-110042
*Department of Mechanical and Automation Engineering, Indira Gandhi Institute of Technology, Delhi
**Department of Training and Technical Education, Delhi

**Abstract**

The gas turbine based power generation plants have been overloaded during summer in India to meet the peak load demand. The present investigation has been carried out to optimize the existing gas turbine based power generation plant using Retrofitting technologies. A Computational analysis of performance parameters has been carried out follower by the exergetic and parametric analysis. The parametric study has been carried out to investigate the effect of ambient temperature and relative humidity on various performance parameters (net poweroutput, thermal efficiency, power generation efficiency , first law efficiency and exergetic efficiency (second law efficiency), heat rate and energy destruction rate). Energy analysis showed that combustion chamber and turbine are most sensitive components of retrofitted system. The results shows that the retrofitting techniques improves the performance parameters. The ambient temperature and relative humidity play key role to enhance the performance of power generation plant.

***Key words:*** Gas turbine, Energy, FCS, HRSG, STIG, Retrofitting.

## Introduction

Simple gas turbine power generation systems are widely used in Indian industries due to quick startup and shutdown capabilities. These units are mostly used to fulfil the peak load demand but unfortunately suffer from very low overall efficiency and reduction in power output during the summer season, when electricity demand is the highest. To investigate the anticipated power shortage, retrofitting projects have been seriously gaining momentum to covert these existing simple gas turbine systems into relatively advanced cycle units resulting in both improved efficiency and power output.

From the fundamental thermodynamic point of view, the reason for low thermal efficiency of simple gas turbine are high back work ratio (large part of turbine work used to compress the inlet air) and substantial amount of available energy loss due to high temperature exhaust (ofter above $500^0$C) caused by shaft rotation of turbine at a relativelely high back pressure. This waste heat recovery from the gas turbine exhaust can be utilized to improve generation efficiency through various modifications to basic cycle such as gas to gas recuperation, steam injection, chemical recuperation, inlet air cooling and combined cycle etc. Among many well-established technologies, the combined cycle is rated as the most efficient way to recover the energy from the exhaust for boosting the capacity and efficiency of power generation. However, the combined cycle technology is unsuitable for daily on-off operation pattern units due to low mobility (start-up time).

Recently, the steam injection gas turbine (STIG) and inlet air cooling (IAC) by evaporation are the most common practices to enhance the performance of power generation. Both feature can be implemented in the existing basic system without major modification to the original system integration in a economic way. In the STIG technology, the steam generated from the heat-recovery generator (HRSG) is injected into the combustion chamber. Both steam from the HRSG and air from the compressor receive fuel energy in the combustion chamber and both simultaneously expand inside the existing turbine to boost the power output of turbine. It should be noted that the required injected steam pressure is obtained from a pump fitted in the steam unit. The net power output produced by steam is significantly higher than that of air in terms of per unit mass flow rate, due to very small pumping work compared to compressor. The thermodynamic processes of a simple cycle gas turbine can be approximately modeled as a Brayton cycle, in which the back work ratio is usually very high, and the exhaust temperature is often above 500 $^0$C. A high exhaust temperature implies there is plenty of useful energy rejected to the environment. The recovery of this wasted energy can otherwise be used to improve either the power generation capacity or efficiency via retrofitting to the basic cycle such as STIG and inlet air cooling by evaporation technology. Since, the specific heat of steam and hence enthalpy is relatively higher than air at a certain temperature, the STIG method is a very effective alternative to increase the efficiency and boost the power output of gas turbine. Arora et al.[1] have carried out energy analysis of a vapour compression refrigeration system with R-22, R-407C and R-410A. Arora et al.[2] have investigated CFD analysis of a Free Power Turbine for an Auxiliary Power Unit. Arora et al.[3] have carried out performance analysis of R22, R417a for 1.5 TR window A.C. Arora & Rai[4] have carried out optimization of thermal efficiency of combined cycle power generation. Arora & Arora[5] have investigated energy analysis of vapor compression system using R22 and its alternate refrigerant R410a. Arora & Kaushik[6] have carried out theoretical analysis of LiBr/$H_2O$ absorption refrigeration systems. Arora et al.[7] have performed energy and energy analysis of a combined transcritical $CO_2$ compression refrigeration and single effect $H_2O$-LiBr vapour absorption system. Wang & Chiou[8] suggested that application of IAC and STIG technique can boost the output and generation efficiency. They concluded that implementing both STIG and IAC features cause more than a 70% boost in power and 20.4 % improvement in heat rate. Bouam et al.[9] have studied combustion chamber steam injection for gas turbine performance improvement during high ambient temperature operations. Their research study is to improve the principal characteristics of gas turbine used under extreme ambient condition in Algerian Sahara by injecting steam in the combustion chamber. Srinivas et al.[10] have worked out on sensitivity analysis of STIG based combined cycle with dual pressure HRSG. They concluded that steam injection decreases combustion chamber and gas reheater exergetic loss from 38.5 to 37.4% compared to the case without steam injection in the combustion chamber. Minciuc et al.[11] have presented thermodynamic analysis of tri-generation with absorption chilling machine. They have focused on solutions of tri-generation plants based on gas turbine or internal combustion engine with absorption chilling machine. Moran [12] has developed design and economic methodology for the gas turbine cogeneration system. Nishida et al.[13] have analyzed the performance characteristics of two configuration of regenerative steam-injection gas turbine (RSTIG) systems. They concluded that the thermal efficiencies of the RSTIG systems are higher than those of regenerative, water injected and STIG systems and the specific power is larger than that of regenerative cycle.

The IAC technology is simply to cool down the inlet air entering the compressor with a cooler. Due to this, the compressor consumes less work and can compress more air per cycle to increase the capacity of the gas turbine. Different IAC options are evaporative cooling, mechanical chiller, absorption chiller and thermal energy storage, etc. applied in gas turbine power augmentation. Among them evaporative cooling is the cheapest one. Sinha & Bansode[14] have studied the effect of fog cooling system for inlet air cooling. They concluded that performance parameters indicative of inlet fogging effects have a definite correlation with the climate condition (humidity and temperature) and showed improvement in turbine power and heat rate with inlet fogging. Chaker et al.[15] have developed the formulation for fog droplet sizing analysis and discussed various nozzle types, measurement and testing. This study describes the different available measurement techniques, design aspects of nozzles, with experimental and provides recommendations for a standardized nozzle testing method for gas turbine inlet air fogging. Salvi & Pierpaloli[16] have studied optimization of inlet air cooling systems for steam injected gas turbines. They proposed the technique of compression inlet air cooling through an ejection system supplied by the exhaust heat of the gas turbine. Bassily[17] has studied the performance improvements of the intercooled, reheat and recuperated gas turbine cycle using absorption inlet-cooling and evaporative after-cooling. A parametric study of the effect of pressure ratio, ambient temperature, ambient relative-humidity, turbine's inlet-temperature (TIT), and the effectiveness of the recuperated heat-exchanger on the performance of varieties of cycles is carried out. Bhargava & Homji[18] have studied parametric analysis of existing gas turbines with inlet evaporative and overspray fogging. This study shows the effects of inlet fogging on a large number of commercially available gas turbines.

Although many efforts have been focused to apply either STIG technology or the IAC method to enhance the gas turbine's performance, very limited studies are available regarding simultaneous integration of STIG and IAC for the same system. In this study, a simple cycle generation unit is carried as base data and STIG and IAC features are retrofitting by added in a sequencing of the system. The benefits obtained from either the STIG or IAC can be distinguished or the integration effect from the combined STIG and IAC can be realized. The performance analysis has been carried out by varying ambient temperature, humidity ratio and steam injection ratio.

**Nomenclature**

| | |
|---|---|
| AP | Approach point ($^0$C) |
| $\dot{E}$ | Energy rate (kW) |
| $\dot{m}$ | mass flow rate (kg/s) |
| M | Molecular weight (kg/kmol) |
| PP | Pinch point($^0$C) |
| TIT | Turbine Inlet Temperature (K) |
| U | Internal energy (kJ) |
| wbt | Wet bulb temperature($^0$C) |

**Superscript**

| | |
|---|---|
| , | fraction of gas phase at dead sta |
| 1 | fraction of gas phase before combustion |
| CH | chemical |
| PH | Physical |
| PT | potential |

**Subscripts**

| | |
|---|---|
| sup | superheated |
| Th | thermal |
| HRSG | Heat recovery steam generator |
| GEN | Heat generator |
| f | fuel |
| REG | Regenerator |
| dbt | Dry bulb temperature($^0$C) |
| CCh. | Combustion Chamber |

**Greek symbol**

| | |
|---|---|
| $\eta$ | efficiency |
| $\lambda$ | fuel-air ratio |
| $\omega$ | steam-injection ratio |
| € | Exergetic efficiency |

**Acronyms and abbreviations**

| | |
|---|---|
| FCS | Fog cooling system |
| STIG | Steam injection gas turbine |

### System Description

The simple cycle gas turbine system integrated with both IAC and STIG featuring are shown in Figure 1. The basic unit includes compressor, combustor, gas turbine and a generator. A heat recovery steam generator (HRSG) was installed at the downstream exit of the turbine (state point 5) in order to recover the heat from the exhaust gases. The fraction of superheated steam generated from the HRSG is used for STIG (state point 9) and the remaining steam is used for process application. An evaporative fog cooling system (FCS) is installed to cool the ambient air (state point 1′) as shown in Fig. 1. Fog cooling is an active system which uses very fine fog droplets of high pressure water injected through special atomizing nozzles located at discrete points across the inlet duct at high pressure to create the cooling effect. The amount of fog is to be monitored base on dry and wet bulb ambient conditions to achieve the required cooling. A typical fog cooling system consists of a high pressure pump skid connected for feeding to an array of manifolds located at a suitable place across the compressor inlet duct. The manifolds have a requisite number of fog nozzles[6] which inject very fine droplets of water into the inlet air. The discharge through each nozzle is around 3ml/s and produces 3 billion droplets per second. The fine fog evaporates very fast, thus dropping inlet air temperature.

*Simple Cycle Gas Turbine with Fog Cooling and STIG*

Simple cycle gas turbine with fog cooling system and STIG consists of a compressor, a combustion chamber, a turbine and a HRSG (heat recovery steam generator) as major components along with a fog cooling system and an STIG mechanism as shown in figure 4.5

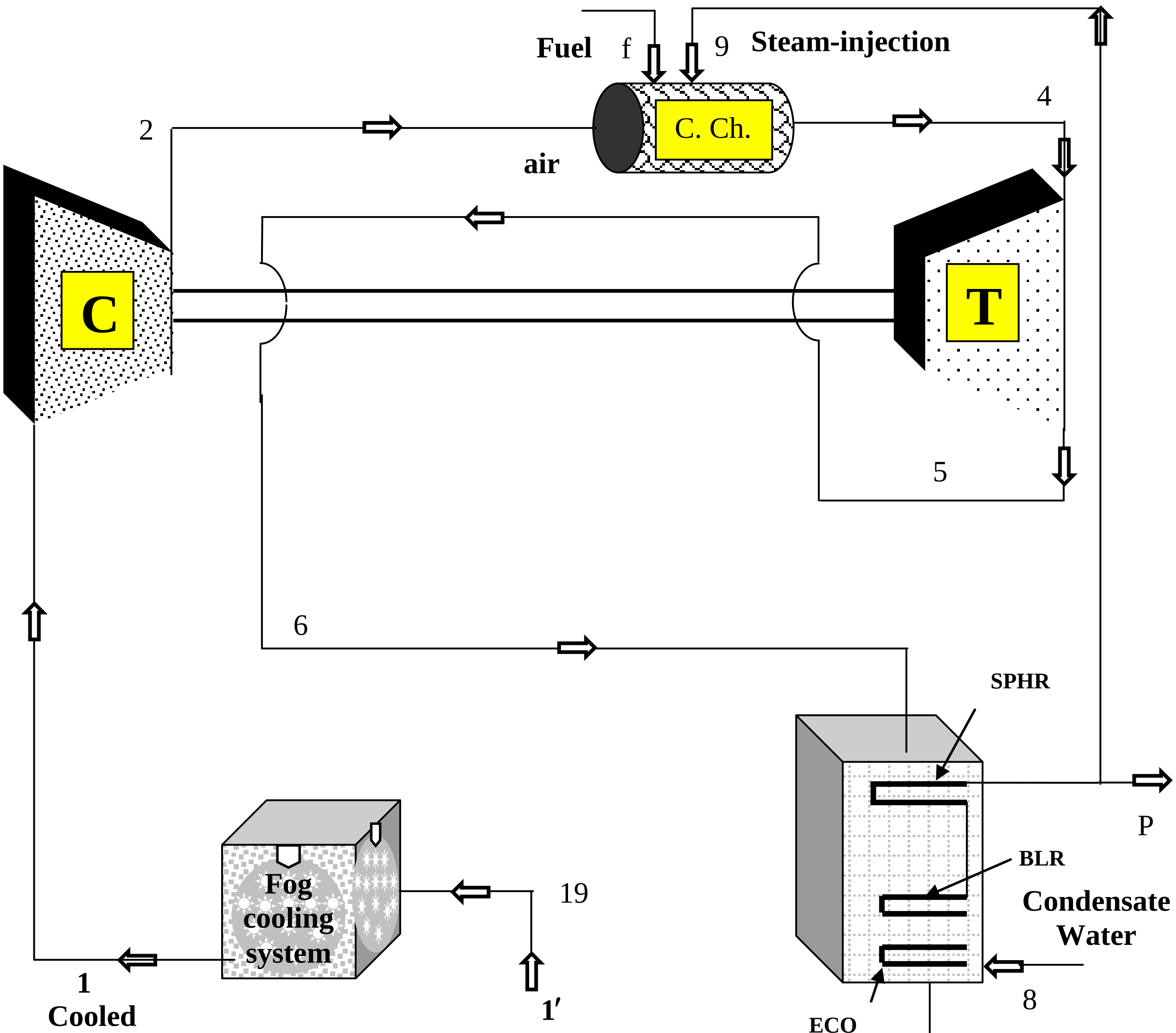

**Fig: 1- Simple gas turbine with fog cooling system and STIG system**

## Modeling and Computer simulation

### Formulation

The following assumptions have been considered for the present study:

1. The composition of dry air has been assumed in terms of molar fraction of 1mole of air is: ($N_2$ = 0.78981, $O_2$ = 0.20989, $CO_2$ = 0.00031 and $H_2O$ = 0).
2. The heat loss from the combustion chamber is 2% of the fuel lower heating value. All other components operate without heat loss.
3. Fog cooling system has been maintained for 100% saturation of ambient air at wet bulb temperature of air.
3. The pressure of water injected from the nozzle into the evaporative cooling chamber has been assumed 138 bar and converts into the fog (fine droplets), absorbs latent heat of air through adiabatic mixing.
4. Combustion chamber has been maintained at constant temperature in the presence of steam (in case of steam injection).

A computer program has been developed to formulate and simulate the retrofitting techniques over simple gas turbine with a set of steady-state governing equations including mass, energy, entropy and energy balances using control volume analysis sequentially for compressor, combustor, gas turbine and HRSG.

For complete combustion of natural gas (methane) with steam injection in the combustion chamber, chemical equation takes the following form

$$\lambda CH_4 + \left[X_{1,N2}N_2 + X_{1,O2}O_2 + X_{1,CO2}CO_2 + X_{1,H2O}H_2O\right] + wH_2O$$
$$\rightarrow \left[1+\lambda+w\right] \quad \left[X_{2',N2}N_2 + X_{2',O2}O_2 + X_{2',CO2}CO_2 + X_{2',H2O}H_2O\right] \qquad (1)$$

Mole fraction of $N_2$ $\quad X_{2,N2} = \dfrac{X_{1,N2}}{1+\lambda+w}$ (2)

Mole fraction of $O_2$ $\quad X_{2,O2} = \dfrac{X_{1,O2} - 2\lambda}{1+\lambda+w}$ (3)

Mole fraction of $CO_2$ $\quad X_{2,CO2} = \dfrac{X_{1,CO2} + \lambda}{1+\lambda+w}$ (4)

Mole fraction of $H_2O$ $\quad X_{2,H2O} = \dfrac{X_{1,H2O} + 2\lambda + w}{1+\lambda+w}$ (5)

where $w$ is the steam injection ratio defined as the ratio of mass of steam injected to the mass of air supplied. $\omega = \dot{m}_s / \dot{m}_a$ , $\omega' = \dot{m}_s / \dot{m}_g$ , $\omega' = \omega/(1+\lambda)$, $\omega'' = \dfrac{\dot{m}_s}{\dot{m}_f}$ , $\quad \omega'' = \dfrac{\omega}{\lambda}$ (6)

where ω is the mass of steam injected to the mass of air supplied, $\omega'$ is the ratio of mass of steam injected to the mass of combustion gases formed and $\omega''$ is the ratio of mass of steam injected to the mass of fuel supplied[2]. The maximum amount of permitted STIG is 20% of mass flow rate of inlet air[2].

The heat transfer between exhaust gases and condensate water has been taken place in water heat recovery boiler where superheated steam is generated.

$$m_{exh}(h_6 - h_7) = m_w(h_{sup} - h_{cond}) \tag{7}$$

where $m_{exh}$ and $m_w$ are mass flow rate of exhaust gases of turbine and condensate water, $h_6$, $h_7$, $h_{sup}$ and $h_{cond}$ are enthalpies of exhaust gases at state 6 and 7, super-heated steam and condensate water.

$$T_{PP} = T_{sat} + PP \tag{8}$$

$$T_{AP} = T_{sat} - AP \tag{9}$$

where $T_{pp}$ , $T_{sat}$ and $T_{AP}$ are Pinch point temperature, saturation temperature of water and approach point temperature. PP is the pinch-point difference and AP is the approach point difference from saturation temperature.

The temperature of air after fog cooling can be obtained from an energy balance on the dry air, water spray and air-born water vapour before and after the system. Assuming adiabatic mixing, the energy gained by the sprayed water is balanced by the energy lost by the dry air, and the original air-born mixture, after cooling such that:

$$m_w(h_{v1} - h_{w1''}) = m_a(h_{a1'} - h_{a1}) + \omega_{1'} m_a(h_{v1'} - h_{v1}) \tag{10}$$

where $m_w$ and $h_{w1''}$ are the mass flow rate and enthalpy of cooling water , $m_a$ is the mass flow rate of dry air, ($h_{a1'} - h_{a1}$) is the enthalpy change of dry air, ($h_{v1'} - h_{v1}$) is the enthalpy change of water vapour during cooling.

The humidity ratio ($\omega_{1'}$) can be specified as:

$$\omega_{1'} = \frac{0.622 P_{v1'}}{P_{1'} - P_{v1'}} \tag{11}$$

where $P_{v1'}$ is the partial pressure of water vapour and $P_{1'}$ is the total atmospheric pressure. From conservation of mass, the amount of water evaporated is equal to the mass of water vapour at point 1minus the water vapour originally in the air at point $1'$.

$$\Delta m_w = (\omega_1 - \omega_{1'}) m_a \tag{12}$$

where $\omega_1$ is the humidity ratio of air after cooling, The partial pressure of water vapour can be found from the respective relative humidity ($\phi$) by:

$$P_v = \phi P_{sat} \tag{13}$$

where $P_{sat}$ and $P_{sat}$ are the saturation pressures of water for the corresponding temperature. Pressure loss in the adiabatic mixing is neglected.

The enthalpy, entropy, and energy can be determined at each state point using mass and energy balances. The performance parameters required for the thermodynamic analysis of retrofitted systems are given below:

***Thermal Efficiency ($\eta_{Th}$):*** Thermal efficiency of a thermal system is defined as the ratio of net work output ($\dot{W}_{net}$) to the total heat input ($\dot{Q}_f$) of the fuel.

$$\eta_{Th} = \frac{\dot{W}_{net}}{\dot{Q}_f} \tag{14}$$

***Generation Efficiency ($\eta_{Gen}$):*** Generation efficiency of a thermal system is defined as the ratio of electrical power output ($W_{el}$) to the total heat input of the fuel($Q_f$).

$$\eta_{Gen} = \frac{W_{el}}{Q_f} \tag{15}$$

***Heat-Rate (HR in kJ/s/kW):*** Heat rate is defined as the ratio of heat produced by the fuel ($\dot{Q}_f$) to the electrical power output ($\dot{W}_{el}$) of the thermal system.

$$HR = \frac{\dot{Q}_f}{\dot{W}_{el}} \tag{16}$$

***Specific Fuel-Consumption (SFC):*** Specific fuel–consumption of a thermal system is defined as the ratio of mass of fuel to the net work output. It is reciprocal of specific net work ($W_{spec}$).

$$SFC = \frac{\dot{m}_f}{\dot{W}_{net}} \tag{17}$$

***First–Law Efficiency ($\eta_I$):*** The ratio of all the useful energy extracted from the system (electricity and process heat) to the energy of fuel input is known as first-law efficiency. First-law efficiency is also known as fuel utilization efficiency or utilization factor or energetic efficiency. By definition,

$$\eta_I = \frac{\left(\dot{W}_{el} + \dot{Q}_{Pro}\right)}{\dot{Q}_f} \tag{18}$$

where $\dot{Q}_{Pro}$ is process heat rate.

***Second–Law Efficiency ($\eta_{II}$):*** Since energy is more valuable than energy according to the second law of thermodynamics, it is useful to consider both output and input in terms of energy. The amount of energy supplied in the product to the amount of energy associated with the fuel is a more accurate measure of thermodynamic performance of a system, which is called second-law efficiency. It is also called exergetic efficiency (effectiveness or rational efficiency). By definition,

$$\eta_{II} = \frac{\left(\dot{W}_{el} + \dot{E}_{pro}\right)}{\dot{E}_f} \tag{19}$$

where $\dot{E}_{Pro}$ is the energy content of process heat and $\dot{E}_f$ is the energy content of fuel input.

***Energy-Destruction Rate ($\dot{E}_{DR}$)***-The component energy destruction rate can be compared to the total energy destruction rate within the system.

$$\dot{E}_{DR} = \frac{\dot{E}_D}{\dot{E}_{D,tot}} \tag{20}$$

## Results and Discussion

In the present study following configuration with retrofitting have been studied in comparison to simple gas turbine cycle:

(i) Simple gas turbine cycle with inlet air cooling (IAC)

(ii) Simple gas turbine cycle with STIG

(iii) Simple gas turbine cycle with both IAC and STIG.

The initial conditions of these system analysis are as shown in Table 1. In the calculation the steady state operation is investigated without considering the turbine blade cooling. The performance analysis of these retrofitted gas turbine system is done by preparing a computer program in EES and validated with Moran[6]. The temperature, pressure and gas concentration in each component are calculated by taking into consideration of the compositions and proportions of gases and consequently various parameters (refer Table 1) and energy loss of these systems are estimated. The net power output and power generation efficiency are 30MW and 29.93% (shown in Table 3) respectively. Attachment of evaporative coolers with simple cycle slightly improves the performance parameters. Due to reduction in compressor work the net power output increases by 3%. The impact of evaporative cooling will be higher in dry summer season where dry bulb temperature is higher and R.H is lower.

Simple gas turbine cycle with STIG (for steam injection ratio 0.1316) significantly improves the system efficiencies. The net power output increases by 27.4%. Combination of simple cycle with STIG and evaporative cooling further improves the system performance. It is quite obvious that contribution of STIG is quite significant.

- Comparison of simple cycle gas turbine with and without FCS shows that net power output increases by 3.1% and various efficiencies increase by 0.6% while heat rate decreases by 0.6% using FCS technology.
- Comparison of simple cycle gas turbine with and without STIG shows that net power output increases by 27.4% and thermal efficiency increases by 11.4% while heat rate decreases by 10.2% using STIG technology. The power output, power generation efficiency improve appreciably while First-law efficiency and process heat fall with increasing amount of STIG.
- Comparison of simple cycle gas turbine with and without FCS and STIG shows that net power output increases by 30.5% and thermal efficiency increases by 11.59% while heat rate decreases by 10.4% using FCS and STIG combine technology. The generation efficiency and net power output increases while First-law efficiency (utilization factor) and process heat decreases with increasing amount of STIG.

Table 1: Essential input parameters for simple gas turbine cycle and retrofitted systems

| **Parameters** | **Simple gas turbine cycle+Fog cooling+STIG** |
|---|---|
| Ambient air temperature at state 1′,in K | 298.15 |
| Ambient air pressure at state1′, in bar | 1.013 |
| Ambient air relative humidity at state1′, in % | 60 |
| Spray water temperature at state1′, in K | 298.15 |
| Spray water pressure at state1′, in bar | 138 |
| Air inlet pressure to compressor ($P_1$), in bar | 1.013 |
| Air inlet temperature to compressor, ($T_1$) in K | 298.15 |
| Relative humidity of inlet air to compressor at 1, in % | 100 |
| Pressure ratio of compressor ($r_p$) | 10:1 |
| Isentropic efficiency of compressor ($\eta_{SC}$), in % | 0.86 |
| Isentropic efficiency of Turbine ($\eta_{ST}$), in % | 0.86 |
| Lower heating value of fuel (LHV), in kJ/kmol | 802361 |
| Mass flow rate of air, in (kg/s) | 81.4 |
| Turbine inlet temperature (TIT) or maximum cycle temperature ($T_4$), | 1520 |

| | |
|---|---|
| in K | |
| Injection pressure of fuel (methane) ($P_f$), in bar | 12 |
| Injection temperature of fuel (methane) ($T_f$), in K | 298.15 |
| Pressure drop in combustion chamber , in % | 5 |
| Exhaust pressure of combustion products after HRSG ($P_7$), in bar | 1.013 |
| Exhaust temperature of combustion products after HRSG ($T_7$) in K | 403.15 |
| Pressure of steam generation ($P_9$) in bar | 20 |
| Pressure of condensate water at inlet of HRSG ($P_8$), in bar | 20 |
| Temperature of condensate water at inlet to HRSG ($T_8$), in K | 298.15 |
| Pressure drop in HRSG on the gas side, in % | 5 |
| Amount of steam injected , in (% of the mass flow rate of the air) | 10 |
| Temperature of superheated steam STIG ($T_9$), in K | 753.15 |
| Approach point , in K | 2 |
| Pinch point, in K | 20 |

Table 2A: Comparison of operational parameters ( $\dot{m}$ , P, T)

| **State Points** | **Mass flow rate ( $\dot{m}$ ), kg/s** | | | | **Pressure (P) , bar** | | | | **Temperature (T), K** | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | **a** | **b** | **c** | **d** | **a** | **b** | **c** | **d** | **a** | **b** | **c** | **d** |
| **1′** | --- | 81.4 | | 81.4 | --- | 1.013 | | 1.013 | --- | 298.15 | | 298.15 |
| **1′** | --- | 1.14 | | 1.14 | --- | 138 | | 138 | --- | 292.6 | | 292.6 |
| 1 | 81.4 | 82.54 | 81.4 | 82.54 | 1.013 | 1.013 | 1.013 | 1.013 | 298.15 | 292.62 | 298.15 | 292.62 |
| 2 | 81.4 | 82.54 | 81.4 | 82.54 | 10.13 | 10.13 | 10.13 | 10.13 | 604.95 | 594.75 | 604.95 | 594.75 |
| 4 | 83.36 | 84.56 | 91.79 | 92.98 | 9.62 | 9.62 | 9.624 | 9.624 | 1520 | 1520 | 1520 | 1520 |
| 5 | 83.36 | 84.56 | 91.79 | 92.98 | 1.013 | 1.013 | 1.066 | 1.066 | 985.6 | 985.95 | 1006.15 | 1006.15 |
| 7 | | | 91.79 | 92.98 | | | 1.013 | 1.013 | | | 403.15 | 403.15 |
| 8 | | | 21.23 | 21.51 | | | 20 | 20 | | | 298.15 | 298.15 |
| 9 | | | 8.14 | 8.14 | | | 20 | 20 | | | 753.15 | 753.15 |
| P | | | 13.09 | 13.37 | | | 20 | 20 | | | 753.15 | 753.15 |
| F | 1.96 | 2.01 | 2.247 | 2.297 | 12 | 12 | 12 | 12 | 298.15 | 298.15 | 298.15 | 298.15 |

a: Simple cycle b: Simple cycle +FCS c: Simple cycle + STIG d: Simple cycle + FCS + STIG

Table 2B: Comparison of operational parameters (h, s and Energy)

| State points | Enthalpy (h), kJ/kmol | | | | Entropy (s) ,kJ/kmolK | | | | Total Energy, MW | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | a | b | c | d | a | b | c | d | a | b | c | d |
| 1′ | --- | -282.8 + 1702 | | -282.8 + 1702 | --- | 198.3 +5.142 | | 198.3 + 5.142 | --- | 0.4525 | | 0.4527 |
| 1 | -337.32 | -471.69 | -282.84 | -471.69 | 198.8 | 198.3 | 198.8 | 198.3 | 0 | 0.0040 | 0 | 0.0040 |
| 2 | 8982.84 | 8684.98 | 9037.42 | 8684.98 | 201 | 200.5 | 201 | 200.5 | 24.588 | 24.470 | 24.59 | 24.48 |
| 4 | 829.91 | 483.00 | -21286.06 | -21319.47 | 236.03 | 236.05 | 237.52 | 237.52 | 92.952 | 94.445 | 111.32 | 111.95 |
| 5 | -18228.21 | -18577.22 | -40133.06 | -40166.73 | 239.35 | 239.35 | 240.7 | 240.7 | 33.279 | 33.801 | 41.59 | 42.139 |
| 7 | --- | | -60596.55 | -60633.01 | --- | -- | 210.36 | 210.36 | --- | | 2.38 | 2.421 |
| 8 | --- | | 1920 | 1920 | --- | | 6.601 | 6.601 | --- | | 0.0935 | 0.095 |
| 9 | --- | | 61676 | 61676 | --- | | 132.8 | 132.8 | --- | | 10.04 | 10.04 |
| P | --- | | 61676 | 61676 | --- | | 132.8 | 132.8 | --- | | 16.15 | 16.49 |
| f | -74872 | -74872 | -74872 | -74872 | -- | | --- | --- | 101.677 | 104.194 | 116.32 | 118.895 |

a: Simple cycle b: Simple cycle +FCS c: Simple cycle + STIG d: Simple cycle + FCS + STIG

TABLE 3: Comparison of various performance parameters of simple gas turbine cycle and retrofitted cycles

| Performance Parameter | Simple gas turbine cycle | Simple gas turbine cycle with Fog cooling | Simple gas turbine cycle with STIG | Simple gas turbine cycle with Fog cooling & STIG |
|---|---|---|---|---|
| First law efficiency (%) | 30.54 | 30.72 | 72.57 | 72.69 |
| Second law efficiency (%) | 29.51 | 29.70 | 55.3 | 55.2 |
| Power generation efficiency (%) | 29.93 | 30.11 | 33.33 | 33.4 |
| Thermal efficiency (%) | 30.54 | 30.72 | 34.01 | 34.08 |

| | | | | |
|---|---|---|---|---|
| Fuel-air ratio | 0.0431 | 0.04355 | 0.0493 | 0.04967 |
| Steam injection ratio (per kg of mass of fuel) | --- | --- | 0.1316 | 0.13 |
| Heat rate (kW/kWh) | 12029 | 11958 | 10800 | 10780 |
| Specific net work (kJ/kg of fuel) or specific power (kW/kg/s of fuel) | 15274 | 15364 | 17012 | 17043 |
| Specific fuel consumption( kg/kWh) | 0.2357 | 0.2343 | 0.2116 | 0.2112 |
| Work-heat ratio (kJ/kJ) | --- | --- | 0.8823 | 0.8826 |
| Power-to-heat ratio (kW/kJ/s) | --- | --- | 0.8647 | 0.8649 |
| Specific work ISO (kW-s/kg of air) | 361.2 | 367.2 | 460.2 | 464.8 |
| Turbine work (MW) | 56.48 | 57.31 | 64.71 | 65.54 |
| Compressor work (MW) | 26.48 | 26.38 | 26.48 | 26.39 |
| Net Power output (MW) | 30 | 30.93 | 38.23 | 39.15 |
| Electric work done (MW) | 29.4 | 30.31 | 37.46 | 38.36 |
| Process heat (MW) | --- | --- | 43.32 | 44.35 |

| **Components** | **Simple gas turbine cycle** | | | **Simple gas turbine cycle+Fog cooling** | | | **Simple gas turbine cycle+STIG** | | | **Simple gas turbine cycle+Fog cooling+STIG** | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | **Energy destruction rate (MW)** | **€** | **Energy destruction rate (%)** | **Energy destruction rate (MW)** | **€** | **Energy destruction rate (%)** | **Energy destruction rate (MW)** | **€** | **Energy destruction rate (%)** | **Energy destruction rate (MW)** | **€** | **Energy destruction rate (%)** |
| Combustion chamber | 33.31 | 0.67 | 46.47 | 34.22 | 0.67 | 46.42 | 39.633 | 0.69 | 63.89 | 40.66 | 0.68 | 63.72 |

| | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Gas turbine | 3.193 | 0.95 | 4.16 | 3.336 | 0.95 | 4.53 | 5.018 | 0.93 | 8.09 | 5.075 | 0.93 | 7.95 |
| Air compressor or Air compressor assembly | 1.892 | 0.93 | 2.47 | 2.366 | 0.91 | 3.2 | 1.892 | 0.93 | 3.05 | 2.367 | 0.91 | 3.71 |
| HRSG | --- | --- | --- | --- | --- | --- | 13.114 | 0.75 | 21.14 | 13.283 | 0.75 | 20.82 |
| Stack-loss | 33.28 | --- | 46.43 | 33.8 | --- | 45.85 | 2.379 | --- | 3.83 | 2.421 | --- | 3.79 |
| Overall plant ( $\sum \dot{E}_D$ )<br>Power output (MW) | 71.675<br>30 | --- | 100 | 73.722<br>30.93 | --- | 100 | 62.036<br>38.23 | --- | 100 | 63.806<br>39.15 | --- | 100 |
| Total energy destruction per MW of output( $\sum \dot{E}_D / MW$ ) | 2.39 | --- | --- | 2.38 | --- | --- | 1.623 | --- | --- | 1.629 | --- | --- |

Table 4: Comparison of energy destruction in the components for simple gas turbine cycle and retrofitted cycles

*Percentage energy destruction rate is the energy destruction rate within a component as a percentage of the total energy destruction rate within the simple gas turbine system.*

Table 4 represents the comparison of energy destruction rate in the components of simple cycle and retrofitted cycles. The energy destruction rate represents the waste of available energy. Energy destruction of all components have been calculated to enhance the understanding of cycle performance. Table 4 presents the energy destruction of each cycle component after retrofitting. While examine the energy destruction for all components, the combustor has the largest energy destruction and shows the major location of thermodynamic inefficiency because of large irreversibility arising from the combustion reaction and heat transfer. Steam injection will increase the energy destruction due to more mixing and combustion in the combustor. The energy-losses at position 7 (see Fig.1) is considered as energy loss through stack. Since part of exhaust heat is recovered in HRSG, the exhaust energy out of stack can be reduced substantially after retrofitting. The energy losses through stack will not only waste the available energy but also dump the thermal pollution to our living environment.

Exergetic efficiency for each component can be defined as the ratio of $\dot{E}_R$ to $\dot{E}_S$ . Where $\dot{E}_S$ is the energy rate supplied to the component and $\dot{E}_R$ is the energy rate recovered from the component. For a retrofitted cycle with fog cooling and STIG, exergetic efficiencies of compressor, turbine, combustor and HRSG are respectively 91%, 93%, 68% and 75%, among which gas turbine and compressor have a higher exergetic efficiency. This implies most of the energy destruction in compressor and combustor are inexistable. It is interesting to note that although the energy destruction rate of combustor is the highest, energy efficiency of the combustor is higher than that of HRSG. Therefore, there is a greater improvement margin exists for HRSG than for combustor.

- Comparison of simple cycle gas turbine with and without FCS shows that exergetic efficiency gets also improve by 0.64%. However fuel-air ratio increases by 1%. The energy destruction gets reduced into compressor and turbine due to low inlet temperature of air.
- Comparison of simple cycle gas turbine with and without STIG shows that exergetic efficiency gets also improve by 87.4% however fuel-air ratio increases by 14.4%. Energy destruction increases in each system

component except air compressor due to mixing of steam and air. The exergetic efficiency improves appreciably energy destruction rate (%) of combustion chamber falls with increasing amount of STIG.

- Comparison of simple cycle gas turbine with and without FCS and STIG shows that exergetic efficiency gets also improve by 87.06% however fuel-air ratio increases by 15.2%. The energy destruction gets increased in each system component due increasing mass flow rate. Energy destruction rate (%) of system component does not show much variation with relative humidity however with increasing amount of STIG, energy destruction rate of each component increases except combustion chamber and compressor.

*Performance study*

*Simple gas turbine cycle with fog cooling system*

The effect of variation of ambient temperature and relative humidity on performance parameters of simple cycle with FCS has been discussed below.

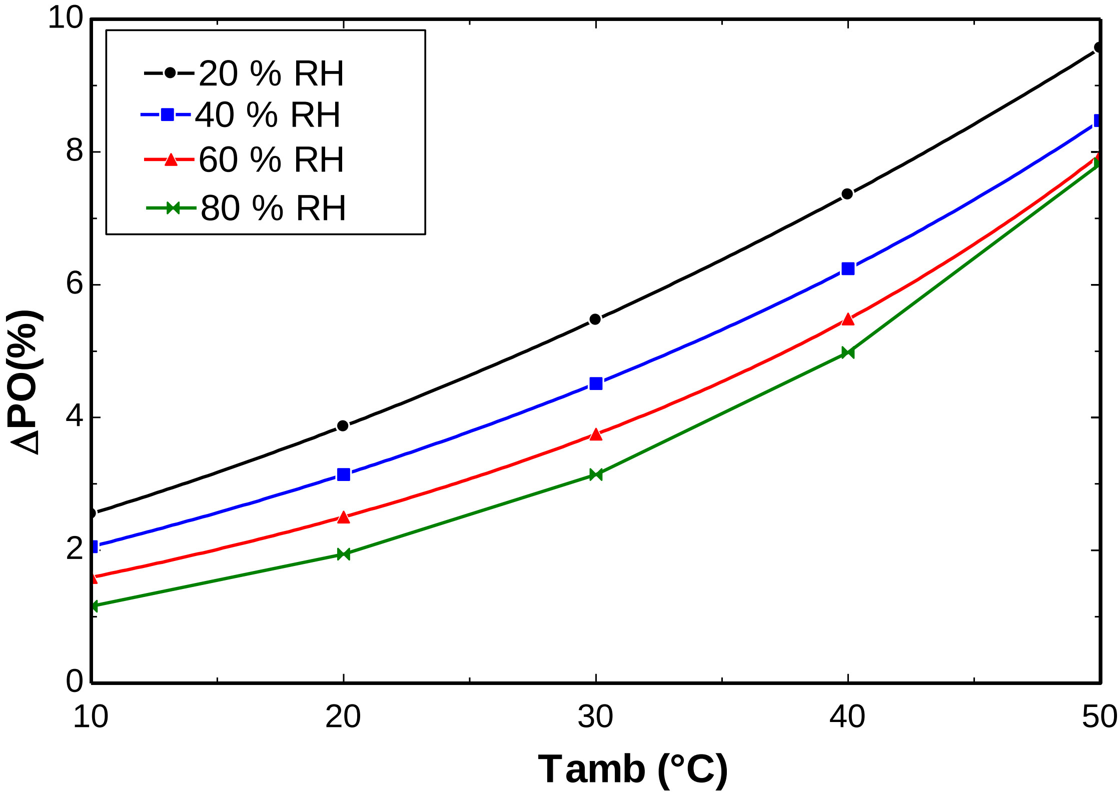


**Fig: 2- Effect of ambient temperature and relative humidity on percentage difference in power output for simple gas turbine cycle with fog cooling**

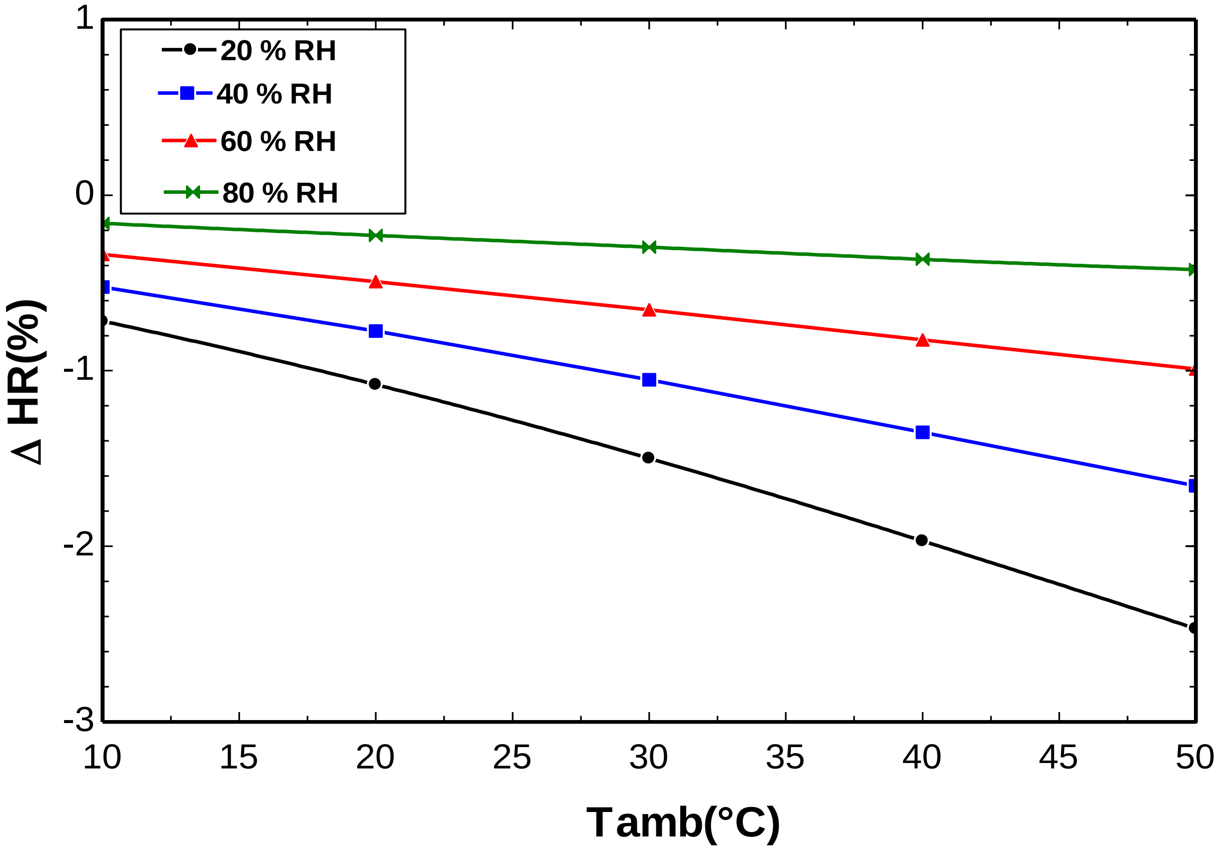


**Fig: 3- Effect of ambient temperature and relative humidity on percentage difference in Heat rate for simple gas turbine cycle with fog cooling**

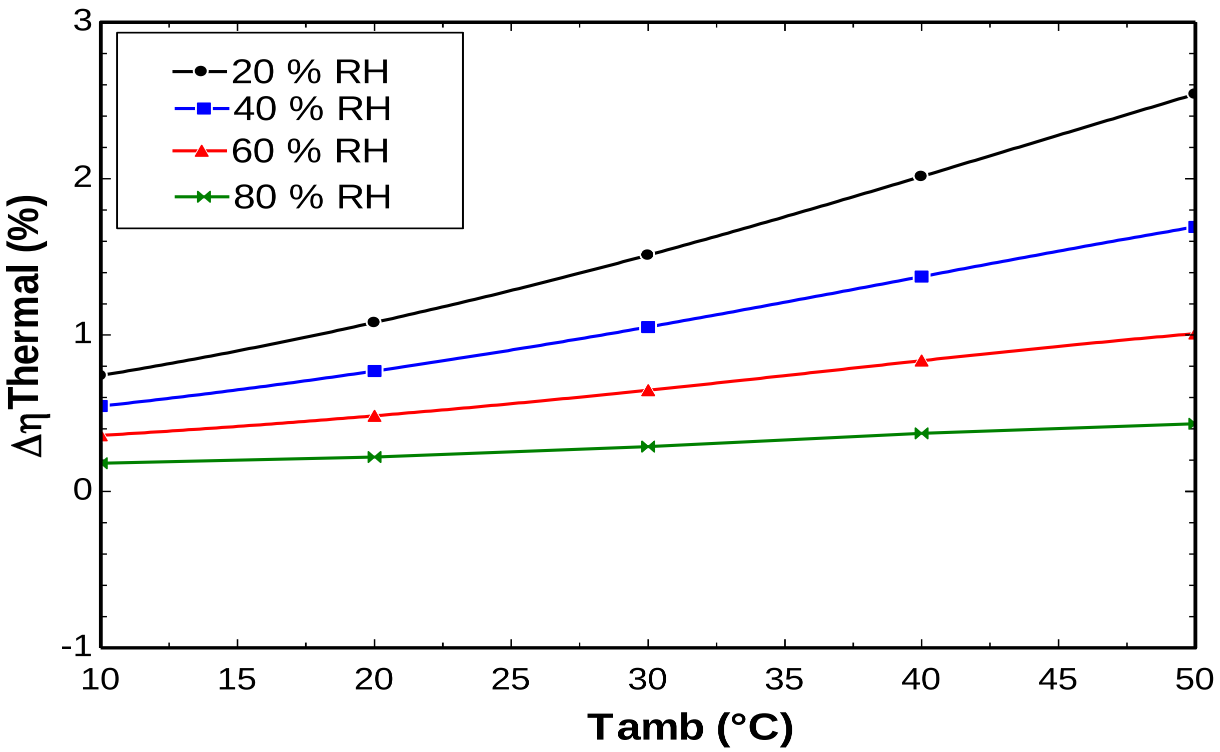

**Fig: 4- Effect of ambient temperature and relative humidity on percentage difference in Thermal efficiency for simple gas turbine cycle with fog cooling**

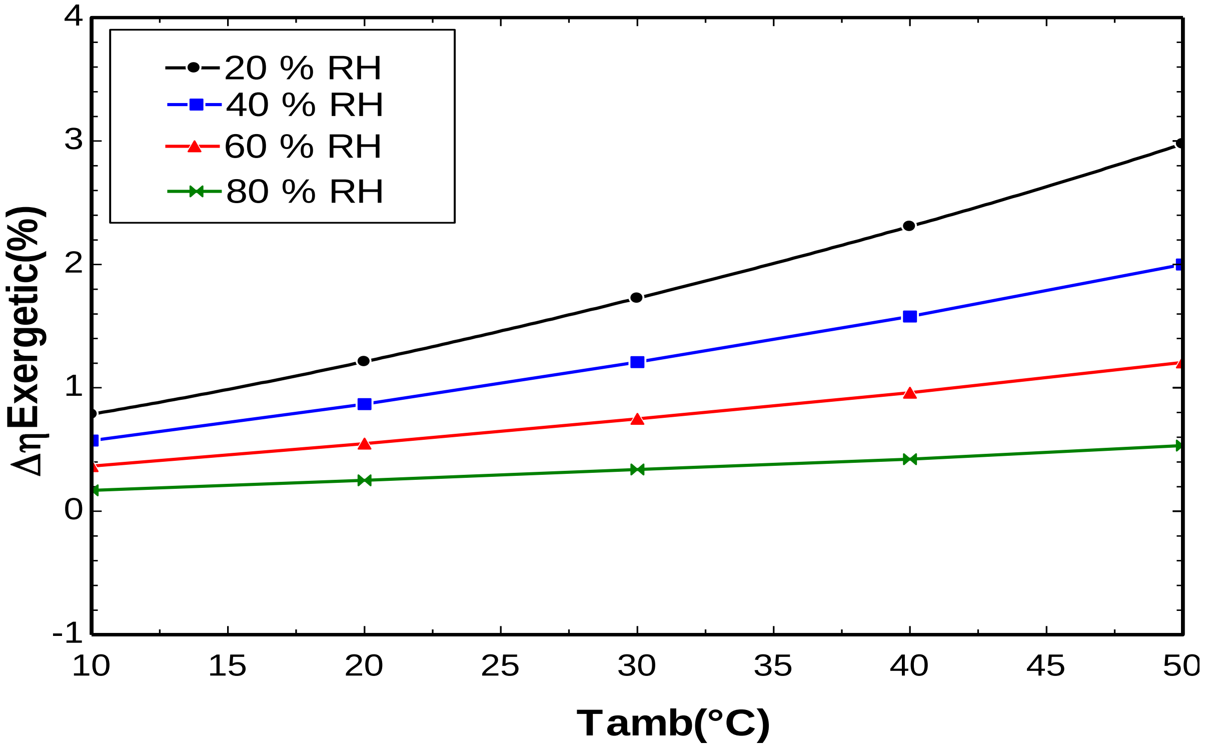


**Fig: 5- Effect of ambient temperature and relative humidity on percentage difference in Exergetic efficiency for simple gas turbine cycle with fog cooling**

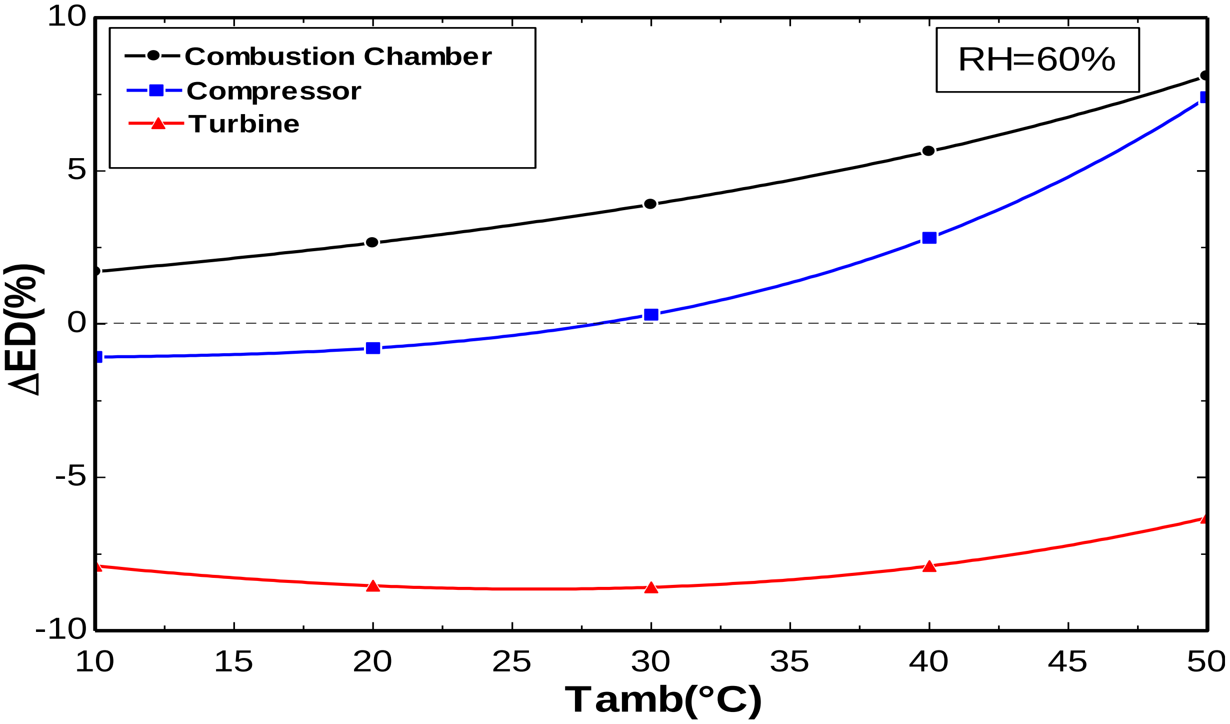


**Fig: 6- Effect of ambient temperature on percentage difference in Energy-destruction for simple gas turbine cycle with fog cooling**

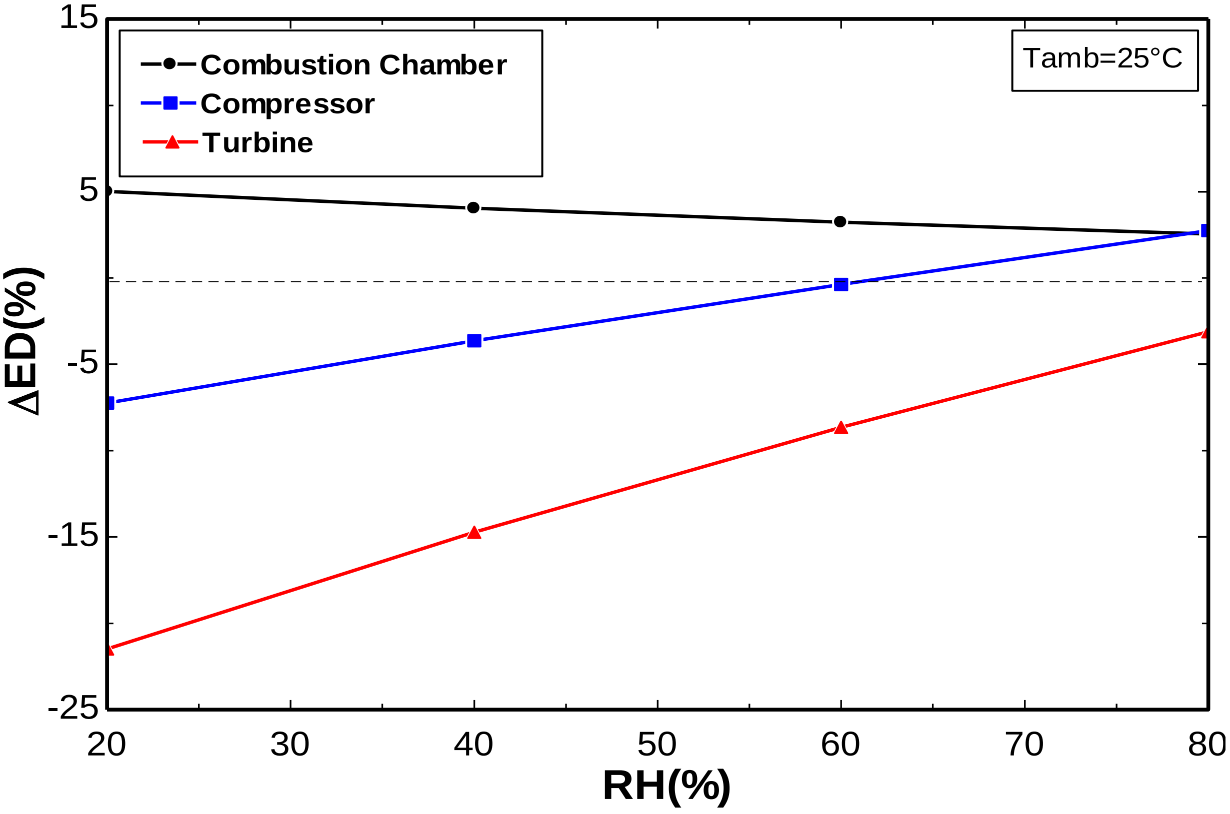


**Fig: 7- Effect of relative-humidity on percentage difference in Energy-destruction for simple gas turbine cycle with fog cooling**

**4.1.1.1 Effect of temperature**

Variation in ΔPO (%), ΔHR (%), $\Delta\eta_{Thermal}$ (%), $\Delta\eta_{Exergetic}$ (%) and ΔηED (%) for simple gas cycle turbine + FCS with temperature at (20-80%) R.H. and 1520 K TIT has been presented in Fig. 3, 4, 5, 6, & 7. Power output increases at higher temperature and lower humidity. Fog cooling can improve 8% relative power output. Use of FCS and vapour contents reduce the air temperature. Therefore, overall effect is to improve the air density and mass flow rate which ultimate reduce the compressor work and improves the net work output. Heat rate decreases at higher temperatures and lower humidity. Thermal efficiency also improves significantly at higher temperature and lower humidity as the net power output increases for the same fuel consumption, the thermal efficiency increases for fog cooling system. The effect of evaporative cooling on exergetic efficiency is higher at high dry-bulb temperature and low relative humidity. For a prescribed temperature and humidity, due to fog cooling the energy destruction rate in compressor and turbine reduces whereas in combustion chamber it increases. With increase in temperature and humidity, the energy destruction increases for all the components.

The trend of graphs shows that the fog cooling with simple cycle gas turbine is a good approach to enhance the performance of the system on the basis of first law. The thermal efficiency, generation efficiency and power output find remarkable improvement while the falling of heat rate is also a good agreement with performance of the system. The enhancement of second law efficiency and fall in energy destruction of combustion chamber also establish that the fog cooling is a performance improvement technique which can be used as a retrofitting technique.

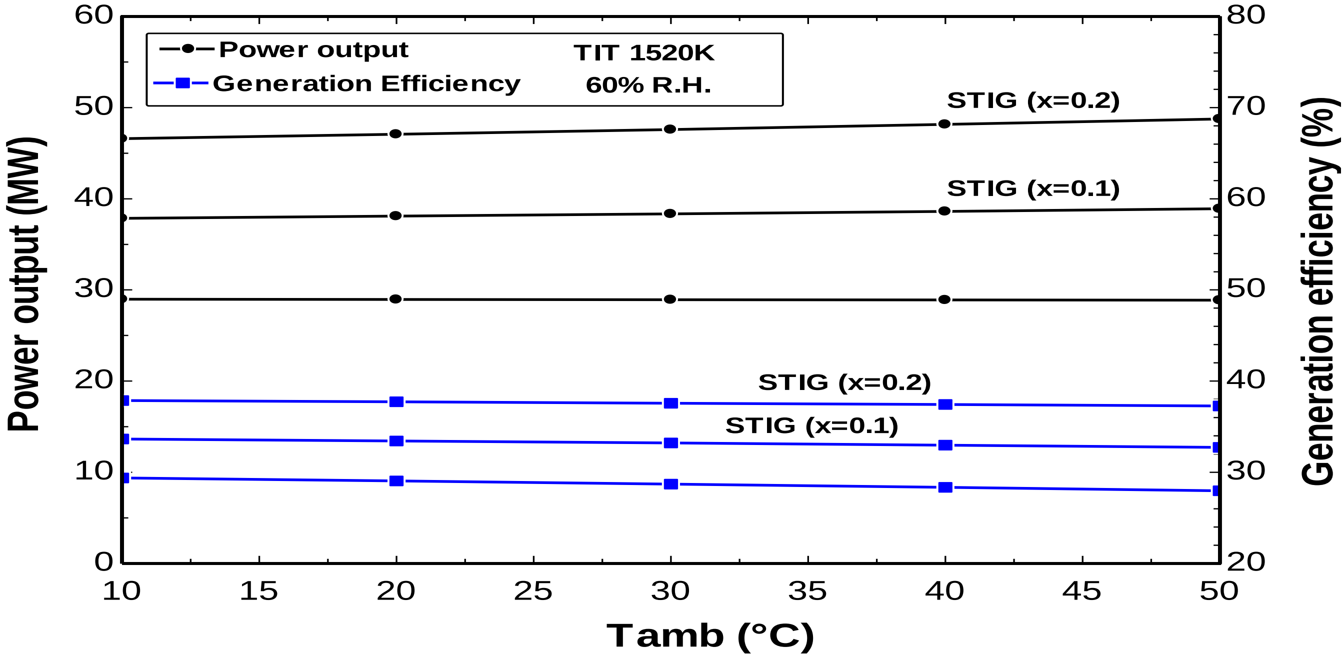


**Fig. 8-The effect of ambient temperatures at power and generation efficiency output**

### 4.1.2 Simple gas turbine with STIG

Various performance parameters have been selected to check the compatibility of simple cycle gas turbine with STIG system. These parameters have been varied with temperature and relative humidity and discussed below. Variation in PO (%) and $\eta_{Generation}$ (%) for simple gas turbine +STIG with temperature at 60% R.H. and 1520 K TIT has been shown in Fig. 9. The power output and generation efficiency increases with the increasing amount of STIG steam for a fixed temperature of inlet air. The power output varies slightly positive with the temperature for a particular trend while the generation efficiency varies slightly negative with the temperature for the same trend. The cause of decrease in generation efficiency is increase in amount of fuel consumption and higher energy destruction in to the combustion chamber with increasing amount of STIG.

Variation in PO (%) and $\eta_{Generation}$ (%) for simple gas turbine +STIG with relative humidity at 25 $^{0}$C ambient temperature and 1520 K TIT has been shown in Fig. 9. The power output and generation efficiency increases with the increasing amount of STIG steam for a particular relative humidity of inlet air. The power output and generation efficiency remains constant with change in relative humidity for a particular temperature.

The effect of STIG on generation efficiency, utilization factor (first law efficiency) and process heat for fixed inlet air condition has been indicated in Fig.11. The utilization factor falls with the increasing amount of steam injection ratio.

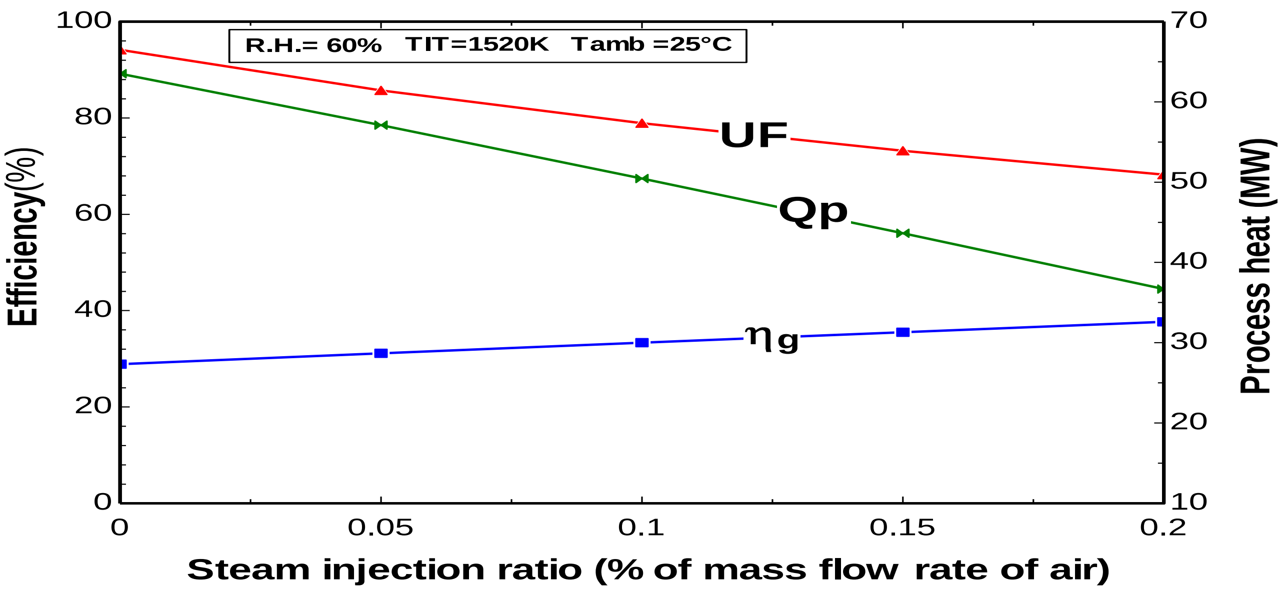


**Fig. 10- The effect of steam injection ratio at utilization factor, generation efficiency and process heat**

The decrease of utilization factor is due to the sharp falling of process heat. The generation efficiency increases while the process heat decreases with increasing amount of steam injection ratio.

The energy destruction rate (%) for different system components with different amount of STIG has been shown in Fig.11. The power output increases for large amount of STIG due to increasing mass flow rate of air. While the energy destruction rate increases into the turbine and HRSG. The energy destruction in combustion chamber is highest among all the system components due to highest temperature of combustion chamber. The graph predicts that steam injection decreases the energy destruction in combustion chamber due to mixing of the combustor lowers the overall temperature.

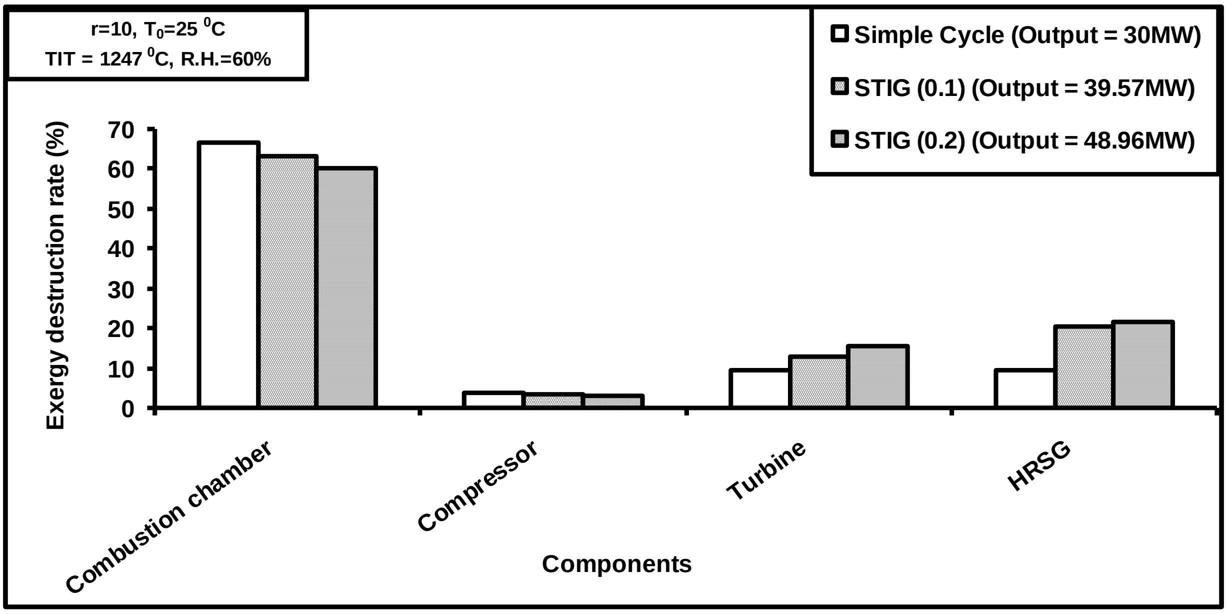


**Fig. 11- Energy destruction rate (%) of system components**

The trend of graphs shows that the STIG with simple cycle gas turbine is best approach to enhance the performance of the system on the basis of first law as well as second law. The thermal efficiency, find remarkable improvement while the falling of heat rate is also a good agreement with the performance of the system. The utilization of waste heat increases the net power output and

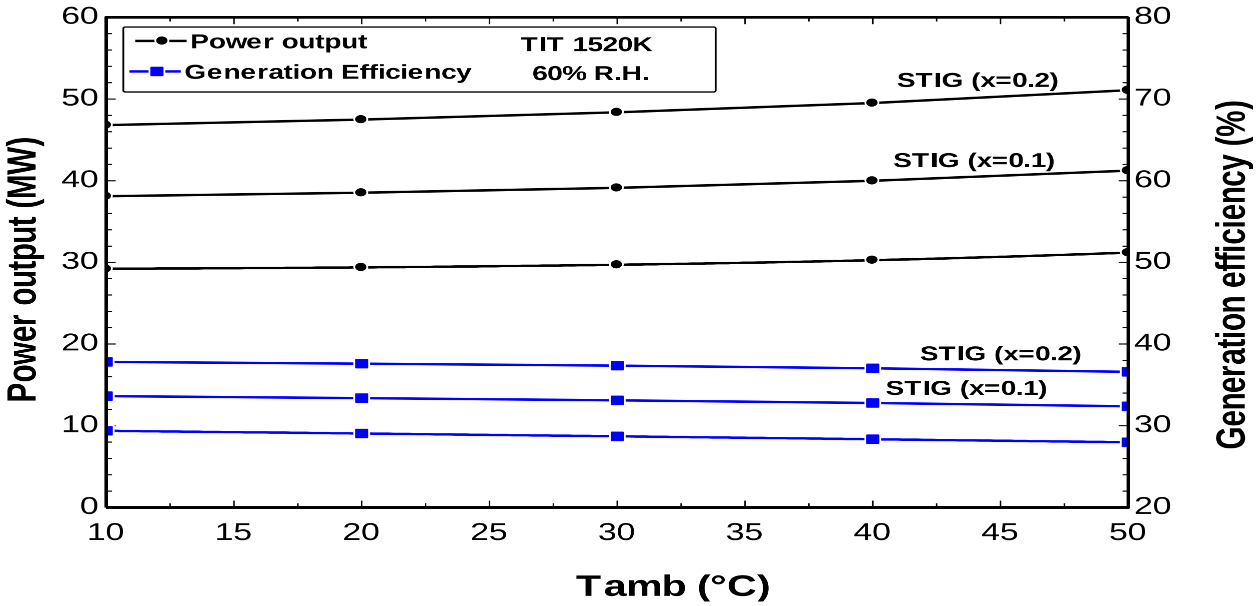


**Fig. 12- The effect of temperature on power output and generation efficiency**

generation efficiency and power output exergetic efficiency while a little amount of fuel consumption increases. The appreciable enhancement of second law efficiency and fall in energy destruction of combustion chamber and compressor also establish that the STIG is a best performance improvement technique which can be used as a retrofitting system.

### 4.1.3 Simple gas turbine with FCS and STIG

Performance parameters have been varied with temperature and relative humidity for simple gas turbine with fog cooling and STIG in order to find effectiveness of combined FCS and STIG retrofitting technique.

The effect of temperature for simple cycle gas turbine +FCS+STIG on PO (%) and $\eta_{Generation}$ (%)at 60% R.H. and 1520 K TIT has been presented in Fig.12 and the air is fogged up to 100% R.H. The power output and generation efficiency increases with the increasing amount of STIG steam for a fixed temperature of inlet air. The power output increases slightly with the temperature for a particular trend while the generation efficiency decreases with the temperature for the same trend. The cause of decrease in generation efficiency is increase in amount of fuel consumption and higher energy destruction in to the combustion chamber with increasing amount of STIG. The gap among power output lines shows that the fog cooling and STIG both enhances the power output of the system.

Variation in PO (%) and $\eta_{Generation}$ (%) for simple gas turbine +FCS+STIG with relative humidity at 25 $^0$C ambient temperature and 1520 K TIT has been shown in Fig.13 while the air gets fogged up to 100% R.H. The power output and

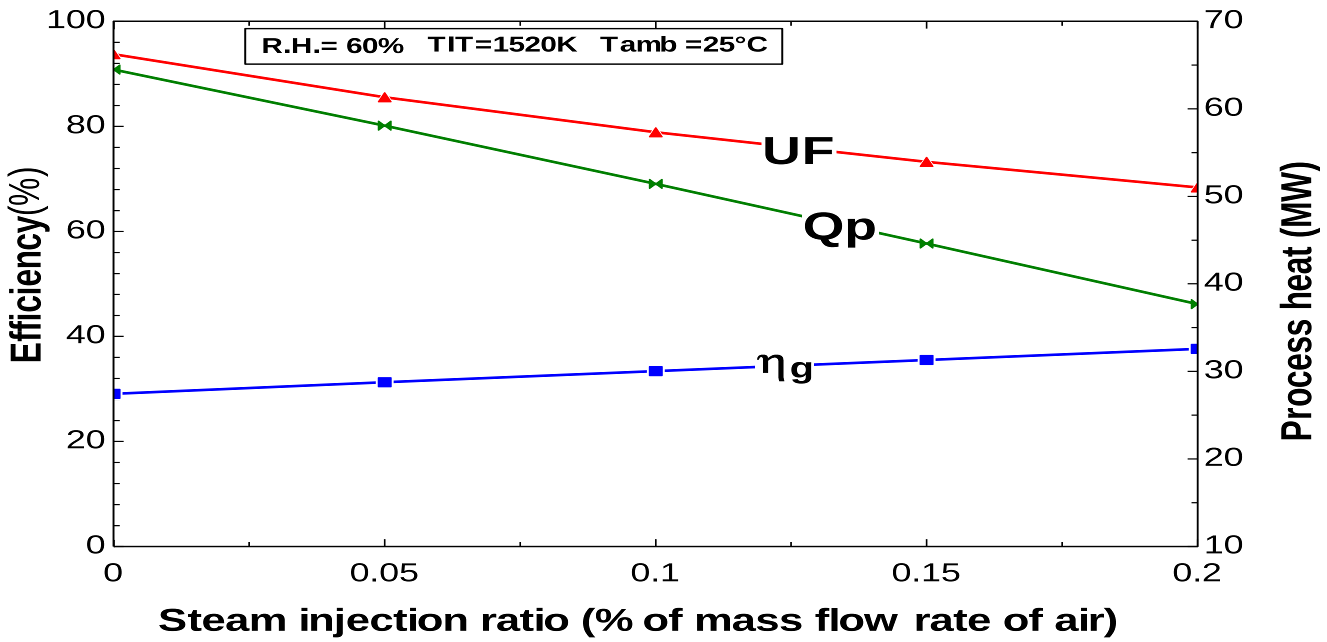




**Fig. 14- The effect of steam injection ratio on utilization-factor, generation efficiency and process heat**

generation efficiency increases with the increasing amount of STIG steam for a particular relative humidity of inlet air along with fogging.

The power output and generation efficiency remains constant with change in relative humidity for a particular temperature along with fogging. The trend o74f graph with varying 41relative humidity is neutral as the temperature of inlet air to the compressor does not change in appreciable amount after fogging.

The Fig. 14 shows the effect of STIG (0-0.2% of mass flow rate of air) on generation efficiency, utilization factor (first law factor is that the slope of process heat is

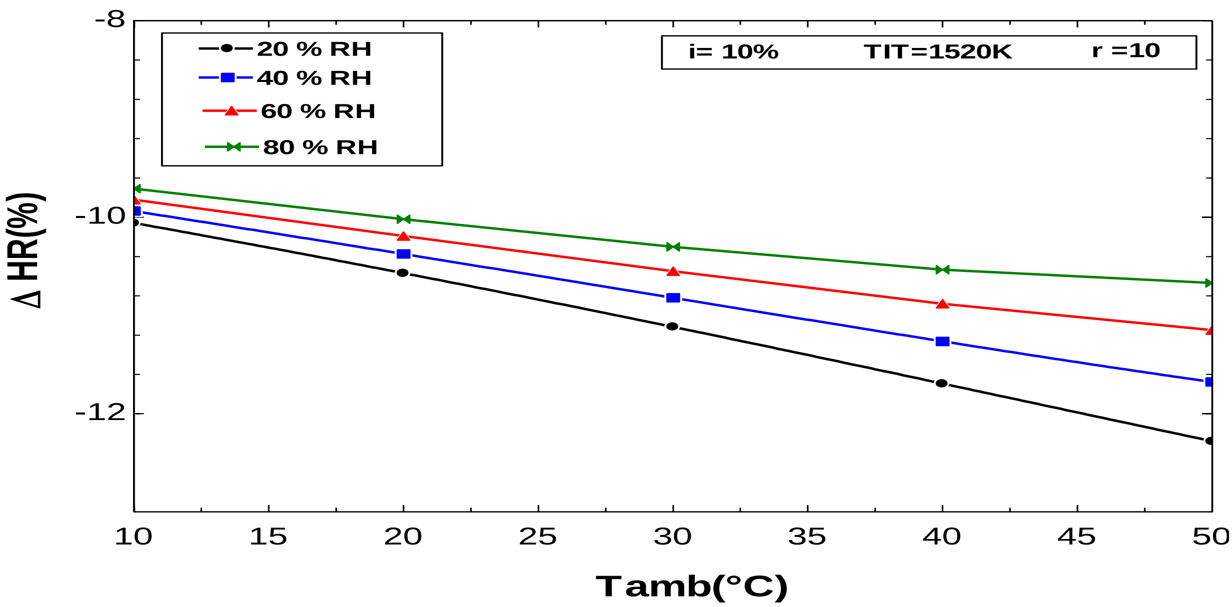


**Fig. 15- The effect of ambient temperature on percentage difference in heat rate**

efficiency) and process heat for fixed inlet air conditions as the air gets fogged up to 100% R.H. The utilization factor falls with the increasing amount of steam injection ratio. The cause of slope of utilization sharper than the slope of generation efficiency. The generation efficiency increases while the process heat falls with increasing amount of steam injection

ratio along with the fogging of air up to 100% R.H. The graph predicts that steam injection effects the generation efficiency, utilization factor and process heat more than the fog cooling of inlet air. Variation of ΔHR (%) for simple gas turbine cycle +FCS+STIG with ambient temperature has been indicated in Fig.15. Percentage difference in heat

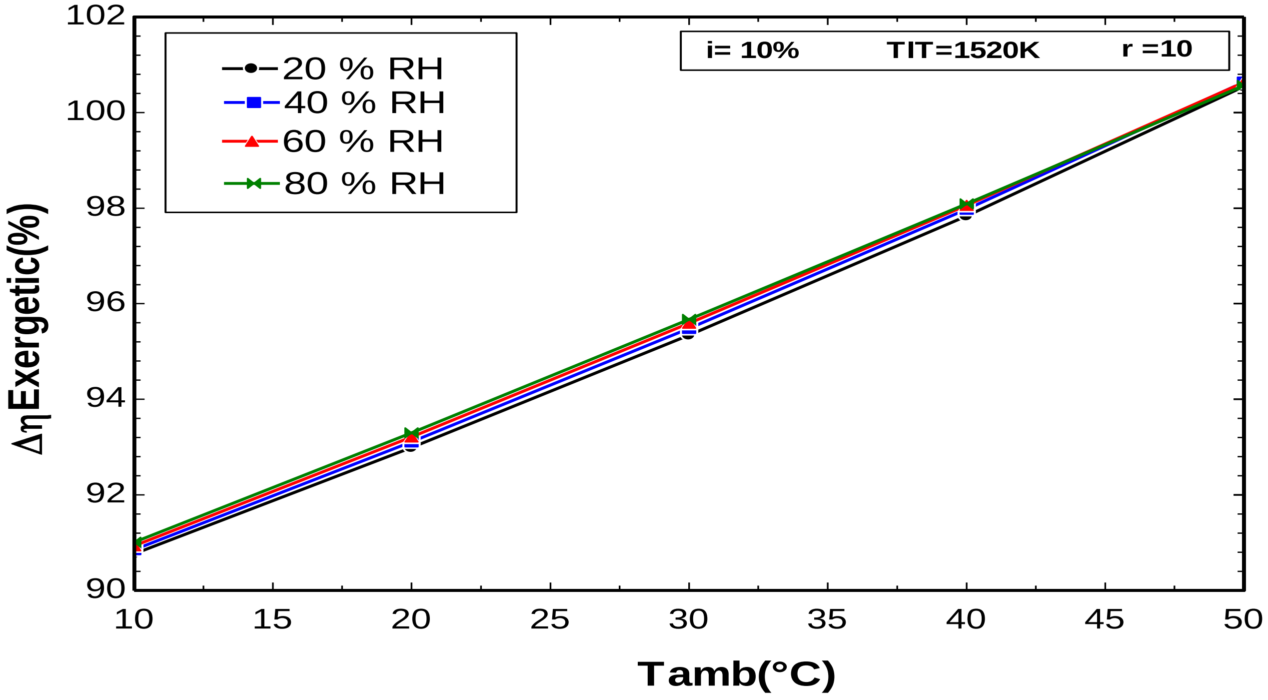


**Fig. 16- The effect of ambient temperature on percentage difference in exergetic efficiency**

rate decreases as the

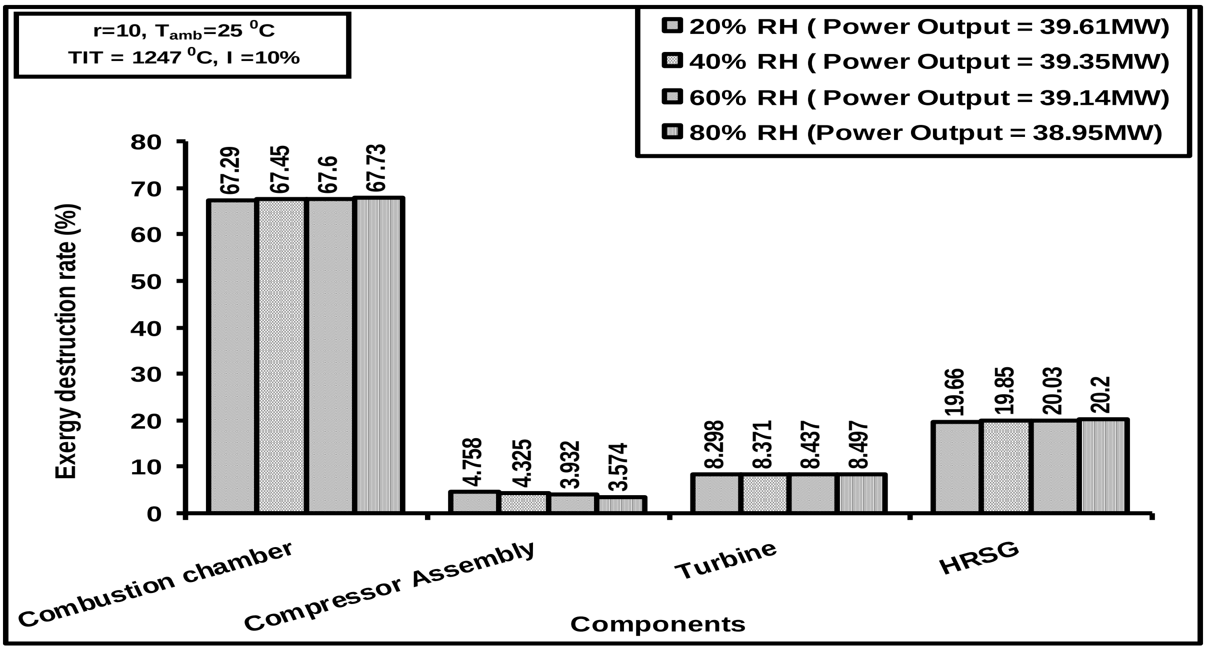


**Fig.17- Energy destruction rate(%) of system components**

temperature increases for a particular relative humidity. If the relative humidity is lower, the percentage difference in heat rate falls rapidly with respect to temperature. In this way fogging with STIG reduces heat rate in appreciable amount as the inlet air gets cooled after fooging.

The Fig.16 indicates that the relationship between percentage difference in exergetic efficiency with ambient temperature of inlet air for different relative humidity as the inlet air gets fogged up to 100% R.H. Percentage difference in exergetic efficiency increases in appreciable amount as the temperature increases for a particular relative humidity. If the relative humidity is lower, the percentage difference in exergetic efficiency is lower for the same trend of temperature. Fogging with STIG impact reverse effect as compare fogging effect in case of energetic efficiency.

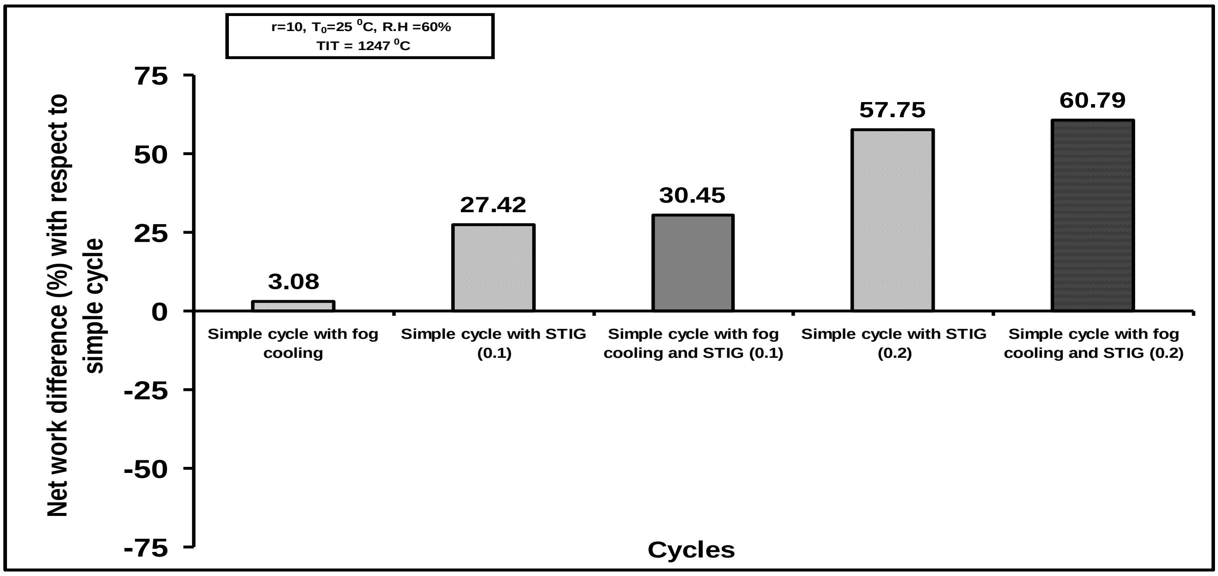


**Fig. 18- Comparison of net work output after retrofitting of**

Percentage energy destruction rate for each system component has been indicated in Fig. 17. It predicts that the power output decreases minutely with increasing relative humidity for a particular amount of STIG. The energy destruction first increases and becomes constant with relative humidity in case of combustion chamber and HRSG while first decreases and becomes constant in case of gas turbine. The energy destruction in the combustion chamber is highest among all the system components due to more temperature existing into the combustion chamber. The graph predicts that fogging and steam injection increases the energy destruction in combustion chamber due to mixing of the combustor raises the overall temperature as the air coming into the combustion chamber is completely saturated.

The trend of graphs shows that the combination of fogging and STIG with simple cycle gas turbine cycle is a good approach to enhance the performance of the system on the basis of first law as well as second law.

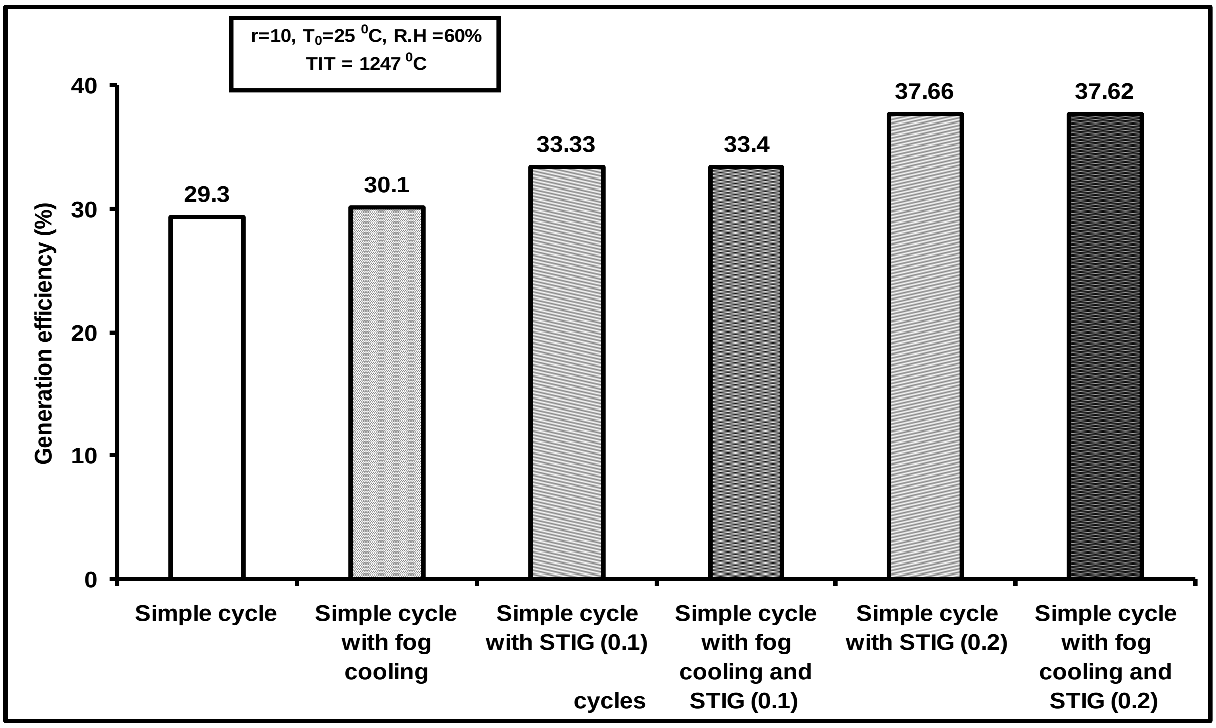


**Fig. 19- Comparison of generation efficiency after retrofitting**

Fig. 18 & 19 predict that power output is maximum in case of simple cycle with fogging and STIG and generation efficiency is maximum for STIG only. The power output increases as the mass flow rate increases and generation efficiency reduces minutely due to higher fuel-air-ratio.

## 5. Conclusions

In this study, the performance characteristics of simple gas turbine retrofitted with either fog cooling system (FCS) or /and steam injected gas turbine (STIG) technology have been analyzed and the various efficiencies and net power output have been investigated, In addition, various performance parameters have also been compared on the basis of first law as well as second law (energy) analysis.

- Comparison of simple cycle gas turbine with and without FCS shows that net power output increases by 3% and various efficiencies increase by 0.6% while heat rate decreases by 0.6% using FCS technology. Exergetic efficiency gets also improve by 0.6%. However fuel-air ratio increases by 1%. The energy destruction gets reduced into compressor and turbine due to low inlet temperature due to low inlet temperature of air. The power output, thermal efficiency and exergetic efficiency enhance with increase in ambient temperature and decrease in R.H. while the heat rate reduces with increasing ambient temperature and decreasing R.H.

- Comparison of simple cycle gas turbine with and without STIG shows that net power output increases by 27.4% and thermal efficiency increases by 11.4% while heat rate decreases by 10.2% using STIG technology. Exergetic efficiency gets also improve by 95% however fuel-air ratio increases by 14.4%. Energy destruction increases in each system component except air compressor due to mixing of steam and air. The power output, power generation efficiency and exergetic efficiency improves appreciably while utilization factor, process heat and energy destruction rate (%) of combustion chamber falls with increasing amount of STIG.
- Comparison of simple cycle gas turbine with and without FCS and STIG shows that net power output increases by 30.47% and thermal efficiency increases by 11.59% while heat rate decreases by 10.4% using FCS and STIG combine technology. Exergetic efficiency gets also improve by 94.4% however fuel-air ratio increases by 15.2%. The energy destruction gets increased in each system component due increasing mass flow rate. The exergetic efficiency enhances while the heat rate reduces with increasing ambient temperature. The generation efficiency and net power output increases while first law efficiency (utilization factor) and process heat decreases with increasing amount of STIG. Energy destruction rate (%) of system component does not show much variation with relative humidity however with increasing amount of STIG, energy destruction rate of each component increases except combustion chamber and compressor.

- Presently the STIG technology is applied only to the gas turbine cycle. It can be extended to the combined power cycle which are now well established. The present study can be further extended for thermoeconomic or exergoeconomic analysis.

**References**


1. Arora A., Arora B.B., Pathak B.D., Sachdev H. L. "Energy Analysis of a Vapour Compression Refrigeration system with R-22, R-407C and R-410A," International Journal of Energy, Vol. 4, No. 4, (2007). Pp..441 -454.

2. Arora B. B., Sharma R. K., Gogoi A., Vipin and Rai J.N., "CFD Analysis of a Free Power Turbine for an Auxiliary Power Unit," International Journal of Theoretical and Applied Mechanics, Volume 5 Number 2 (2010) pp. 223-232.

3. Arora B. B., Arora A., Sachdev H. L., "Performance analysis of R22, R417a for 1.5 TR window A.C," National conference on advances in Mechanical Engineering held at Govt. College, Kota, 18-19 March, 2006. Paper No. ECKAME2006-T-17.

4. Arora B. B. & Rai J. N., "Optimization of Thermal Efficiency of Combined cycle power generation" was presented and published in the National Seminar on Energy & Environment held at Aanad Engineering College, Agra (21-22 Dec. 2001).

5. Arora B. B., Arora A., Sachdev H. L., "Energy analysis of Vapor compression system using R22 and its alternate refrigerant R410a," National conference on advances in Mechanical Engineering held in Jamia Millia Islamia Delhi in Jan 20-21, 2006. p.p.810-817.

6. Arora A., Kaushik, S.C, "Theoretical analysis of LiBr/$H_2O$ absorption refrigeration systems", International Journal of Energy Research 33 (2009) 1321-1340.

7. Arora A., Singh N. K., Monga S. & Kumar O., "Energy and energy analysis of a combined transcritical $CO_2$ compression refrigeration and single effect $H_2O$-LiBr vapour absorption system", International Journal of Energy Volume 9 - Issue 4(2011) 453-470.

8. Wang F.J., Chiou J.S., "Integration of steam injection and inlet air cooling for a gas turbine generation system", Energy conversion and Management 45 (2004) 15-26, Taiwan, ROC.

9. Bouam A., Aissani S., Kadi R., "Combustion chamber steam injection for gas turbine performance

improvement during high ambient temperature operation", Journal of engineering for gas turbines and power, (2008), vol. 130, 041701-10.

**10**.Srinivas T., Gupta A.V.S.S.K.S., Reddy B.V., "Sensitivity analysis of STIG based combined cycle with dual pressure HRSG", International journal of Thermal Science (2007), India.

**11**.Minciuc E., LeCorre O., Athanasovici V., Tazerout M., Bitir I., "Thermodynamic analysis of tri-generation with absorption chilling machine", Applied Thermal Engineering 23 (2003) 1391-1405, France.

**12.**Moran M.J., "Thermal system design and optimization", (John Wiley & Sons, New York) 1996, 156-193.

**13.**Nishida K., Takagi T., Kinoshita S., "Regenerative steam-injection gas turbine systems", Applied Energy 81 (2005) 231-246, Japan.

**14**.Sinha R., Bansode S., "A thermodynamic analysis for gas turbine power optimization by fog cooling system", 20th national and 9th International ISHMT-ASME heat and mass transfer conference, (2010).

**15**.Chaker M., Meher-Homji C.B., Mee III T., "Inlet fogging of gas turbine engines-PartII: fog droplet sizing analysis, nozzle types, measurement, and testing", Journal of engineering for gas turbines and power, (2004), vol. 126/559, Monrovia.

**16**.Salvi D., Pierpaoli P., "Optimization of inlet air cooling systems for steam injected gas turbines", International journal of thermal science 41 (2002) 815-822, Italy.

**17**.Bassily A.M., "Performance improvements of the intercooled reheat recuperated gas turbine cycle using absorption inlet-cooling and evaporation after-cooling", Applied Energy 77 (2004) 249-272, Libya.

**18**.Bhargava R., Meher-Homji C.B., "Parametric analysis of existing gas turbines with inlet evaporative and overspray fogging", Journal of Engineering for gas turbines and power, (2005), Vol. 127/145, Houston.

**19**.Wang J, Zhao P, Niu X, Dai Y, "Parametric analysis of a new combined cooling, heating and power system with transcritical CO2 driven by solar energy", Applied Energy, 94 (2012) 58-64.

**20**.Haebi H, Saidi M H, Ahmadi P, "Exergoeconomic optimization of a trigeneration system for heating, cooling and power production purpose based on TRR method and using evolutionary algorithm", Applied Thermal Engineering, 36 (2012) 113-125.

**21**.Suamir I, Tassou S A, “Performance evaluation of integrated trigeneration and CO2 refrigeration systems”, Applied Thermal Engineering, xxx (2012)1-9.

**22**.Arvalho M, Lozano M A, Serra LM, “Multicriteria synthesis of trigeneration systems considering economic and environmental aspects”, Applied Energy, 91 (2012) 245-254.

**23**.Bravo H E G, Rodriguez R D, Gutierrez J H, Brown R B Y, Aguila R R, Peria H T, “State of art of simple and hybrid jet compression refrigeration systems and the working fluid influence, Int. Journal of Refrigeration”, 35 (2012) 386-396.

**24**.Suamir I, Tassou S A, Marrriott D, “Integration of CO2 refrigeration and trigeneration systems for energy and GHG emission saving in supermarkets”, Int. Journal of Refrigeration, 35 (2012) 407-417.

**25**.Song Y, Wang J, Dai Y, Zhou E, “Thermodynamic analysis of a transcritical CO2 power cycle driven by solar energy with liquefied natural gas as its heat sink”, Applied energy, 92 (2012) 194-203.

**26**.Sha S, Hurme M, “Energy evolution of combined heat and power plant processes”, Applied Thermal Engineering, xxx (2012) 1-8.

**27**.Angrisani G, Rosato A, Roselli C, Sasso M, Sibilio S, “Experimental results of a micro-trigeneration installation”, Applied Thermal Engineering, 38 (2012) 78-90.

**28**.Sevilgen S H, Sancar O, “Economica analysis of trigeneration system, Int. Journal of Physical Sciences”, 6(5) (2011) 1068-1073.

**29**.Reddy K S, Vikram T S H, “Investigations on solar parabolic dish based vapour absorption air conditioning system”, ISHMT-ASME Heat and Mass Transfer Conference, IIT Madras (2011).

**30**.Amaris C, Bourouis M, Valles M, Salavera D, Coronas A, “Thermophysical properties and heat and mass transfer in plate heat exchangers of new absorption mixtures with special reference to solar cooling applications”, ISHMT-ASME Heat and Mass Transfer Conference, IIT Madras (2011).

**31**.Ma S, Wang J, Yan Z, Dai Y, Lu B, “Thermodynamic analysis of a new combined cooling, heat and power system driven by solid oxide fuel cell based on ammonia-water mixture”, Journal of Power Sources, 196 (2011) 8463-8471.