EFFICIENCY AND POWER ENHANCEMENT OF COMBINED CYCLE POWER PLANTS BY INLET AIR CONDITIONING TECHNIQUES

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ABSTRACT

The combined gas/steam turbine cycle power plants offer an efficient and environment friendly system for electric power generation. However the power output from a gas turbine decreases significantly with increase in ambient temperature. This is a serious condition especially in hot climatic environments. Best method to tackle this situation is to condition the inlet air to the gas turbine using innovative techniques for augmenting power and efficiency improvement.

This paper illustrates the influence of various inlet air conditioning techniques. A computational model of the Combined Cycle Gas Turbine (CCGT) plant was developed and applied to a typical 350 MW Combined Cycle Power Plant (CCPP), operational in Kerala, India. The model is based on thermodynamics, heat transfer and psychrometric principles. With the help of this model, a parametric analysis of combined cycle gas turbine plant for various temperatures and humidity with swirl flash technology was carried out and results plotted. Evaporative cooling and refrigerated inlet cooling techniques were compared with the swirl flash technology to establish that the swirl flash is superior to other methods in efficiency and power enhancement.

KEY WORDS

Combined Cycle, Evaporative cooling, Refrigerated inlet cooling, Swirl Flash, Modelling

INTRODUCTION

The Combined gas/steam turbine cycle power plants have long been recognized as an efficient system for electric power generation. Although CCGT plants have a number of advantages like higher efficiency, lower emissions, less lead-time, less space requirement etc they have a serious flaw. Gas turbine power output decreases with increase in ambient temperature, power output dropping by 0.5% to 0.9% for every 1°C rise in ambient temperature. On several heavy frame gas turbines, power output drop to approximately 20% when ambient temperature reaches 35°C, coupled with a heat rate

increase of about 5%. The effects of ambient temperature on the power output and heat rate are shown in figure 1[1]. This problem can be resolved by adopting new innovative methods for power augmentation and efficiency improvements by conditioning the inlet air to gas turbine. Important types of inlet air conditioning methods are discussed in this paper. The innovative method of over spray power augmentation called swirl flash is studied in detail. A software model of CCGT plant operated by NTPC at Kayamkulam is developed and with this model, a parametric study of the swirl flash over spray inlet air conditioner on the CCGT plant is analyzed and findings plotted.
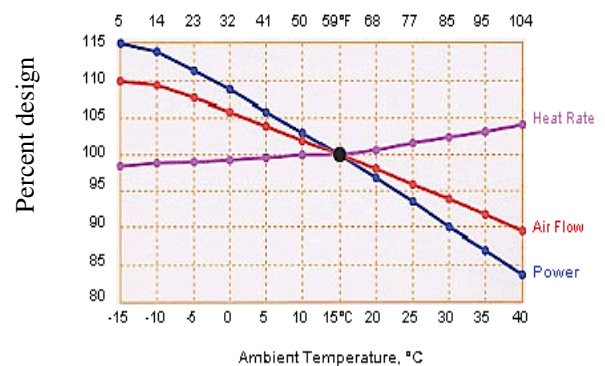
Figure1 Effect of ambient temperature on output and heat rate.

GAS TURBINE INLET AIR COOLING

One of the fiscal ways to maximize power output of a gas turbine is to cool inlet air of gas turbine. The gas turbine's power output increases as air mass flow rate increases provided other variables are kept constant. With the constant volumetric flow of a gas turbine, by increasing the air compressor inlet air density, more mass flow rate is achieved. The power produced by the turbine is nearly a linear function of air mass flow rate. If additional mass flow from the fuel is ignored, then from the

first principles it is clear that the power output is a linear function of temperature. The most common technique utilized in power generation to increase mass flow is to increase the air density by lowering the inlet air temperature.

The available inlet air cooling technologies today are classified either under evaporative or refrigerate cooling systems. Evaporative type systems are viewed to be suitable for hot and dry climates since its cooling capability is limited to the ambient wet bulb temperature. However, it has several advantages such that it requires minor modifications to the inlet plenum, low capital installation cost and it carries a high reliability and low operating and maintenance cost. Refrigeration systems are complex and require much higher capital investment and operating and maintenance cost. However, their applications are not constrained by value of wet-bulb temperature and, thus, higher power augmentation can be achieved. The latest concept of swirl flash technology offer more/superior benefits.

Evaporative Cooling

Evaporative cooling techniques accomplish inlet air temperature reduction through psychrometric processes. Here, wetted media water is brought into contact with the air stream and the warm air transfers heat to the liquid water, evaporating part of the liquid water into vapour. As the air transfers heat to evaporate water, the temperature of the air decreases along the constant wet bulb temperature line (constant enthalpy, without loss or gain of heat). At the same time moisture is added to air.

Refrigerated Inlet Cooling

Here the inlet air passes over a chilled coil, the water vapour content (humidity ratio) remains constant as it decreases in temperature to the saturation curve. If the inlet air is cooled further, the process line follows the saturation curve and water vapour will be condensed out of the inlet air. Mechanical chilling employs the use of refrigerant systems to provide the heat transfer needed to lower the gas turbine inlet air temperature to a consistent entering air temperature somewhat independent of local site conditions. There are centrifugal package chillers and absorption package chillers available for the purpose.

The Swirl-Flash System

The compression of air requires less energy at low temperatures than at high temperatures, because of the smaller volume. In most designs, intercoolers are used to reduce the temperature, but heat exchangers are expensive and should be avoided wherever possible. By spraying water into the compressor and allowing the droplets to evaporate, a similar effect can be achieved. It is not unusual for water to be present in compressors (during cleaning, for example, or in an aircraft turbine, when flying through clouds or showers). However, round-the-clock water injection could cause problems with erosion, water separation etc. The droplets must therefore be small. The challenge is to generate a spray of tiny droplets (typically 1 to 5 µm in diameter) in a quantity sufficient to cool the air during compression. Experiments show that this can best be done by the newly emerged swirl flash technology. The
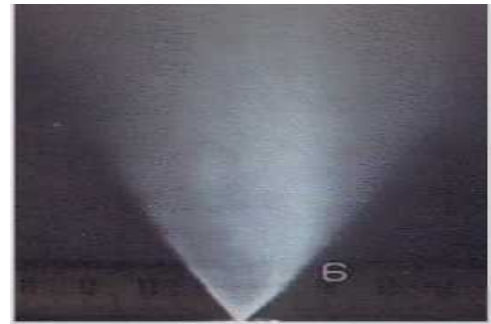

**Figure 2** Spray pattern under cold conditions

Figure 3 Spray pattern in the swirl flash mode

swirl flash technology is based on a simple but robust principle. Water is pressurized (typically 100-150 bar), heated-up to about 200 °C and fed to a swirl nozzle. Due to the swirl movement, the water sprouts out of the nozzle in a typical spray pattern, which has the shape of a cone. The droplets size is about 25 µm. However, when the water is significantly above the boiling point at the ambient pressure, it starts boiling violently (flashing). As a result, each droplet of 25 µm explodes in a thousand fragments, each having the size of about 2.5 µm. See figures 2 and 3 . The typical spray cone of a swirl nozzle changes because of partial flashing to a parabolic shape [9]. The ultra fine spray ensures almost instant evaporation and cooling. The droplet size distribution is indicated below (up to 10^{12} droplets per second!) see figure 4 [9]

The idea of cooling air by adding hot water sounds strange. But the amount of heat extracted from the compressed air by means of evaporation is far greater than the amount of heat added by utilizing the hot water spray. The result is a drop in compressed air temperature and a corresponding drop in compressor discharge temperature. In order to keep the turbine inlet temperature constant, the system has to supply more fuel. In combination with the lower parasitic work for the compressor, this results in a higher output. Single shaft gas
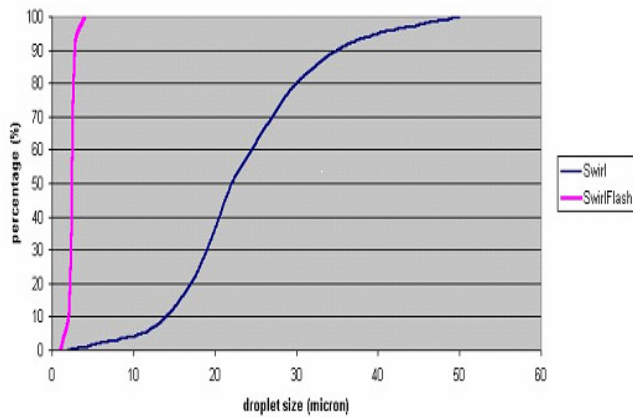
Figure 4 The droplet size distribution in swirl flash nozzle and a swirl nozzle

turbines show a power augmentation of 10%, double shaft turbines can do even more because the shaft of the compressor is not necessarily limited to a fixed number of revolutions.

As a result the compressor can supply (within limits) extra air and the turbine can supply even more power. The compressed humidified air reduces also the stoichiometric adiabatic flame temperature during combustion. This reduces the thermal NOx. For diffusion burners the NOx reduction can be up to 40% and for dry low-NOx burners it is typically 20-25%.[9]

The Swirl Flash technology has a very favourable behaviour when it comes to applicability under various ambient conditions. The classical inlet air chillers can not work properly at high temperatures and low relative humidity. The cold-water over-spray injection systems are limited to ambient temperatures above 10 °C in order to avoid ice formation at high air velocities. The Swirl Flash system, however, can be used in a much wider range. At high temperatures and high humidity, the evaporation takes place almost completely inside the compressor. At high temperature and low humidity the system attains inlet air chilling while the remainder of the water evaporates inside the compressor. At low temperatures (0 °C) and high humidity the hot spray acts as an inlet air de-icing system and can still be used. Only when the humidity of the inlet air is close to zero, the inlet air temperature must be 5 °C in order to avoid ice formation. These features result in a far wider range of temperatures and humidity, where the swirl flash over-spray can be applied. Because of this, the amount of extra-generated power is far greater than any other system.

After examining the various power augmentation techniques it is clear that the swirl flash over spray technique is an emerging technology, which is far superior to others. To examine the effect of this over spray technique on CCGT plant , a typical plant of 350 MW at RGCCPP-NTPC (Rajiv Gandhi Combined Cycle Power Plant - National Thermal Power Corporation), Kayamkulam, Kerala was selected for analysis. A computational model was prepared and parametric analysis carried out to illustrate the performance of swirl flash system.

MODELLING OF THE CCGT PLANT

The design of combined cycle power plants is complicated because of coupling between two different types of power cycles and the need to identify optimal distribution of power production between them. Thus it has become imperative for developing suitable computer simulation techniques that would enable prediction of plant performance at various operating conditions like , with or without inlet air conditioner, different ambient temperature and humidity conditions etc. The plant at RGCCPP-NTPC Kayamkulam, Kerala was selected for modeling. The topping cycle of plant consists of two gas turbines of 115 MW capacity of GE make (frame 9E). The downstream Heat Recovery Steam Generator(HRSG) is unfired, dual pressure units having natural circulation evaporators. The steam turbine is of BHEL (Bharat Heavy Electricals Limited) make having a capacity of 130 MW.

The task of computer simulation involves predicting the operating conditions of the system (pressures, temperatures energy and mass flow rates) at various mass and energy balances, all equations of state of working substances and the performance characteristics of the individual components are satisfied. Therefore, the availability of performance characteristics of the various components constituting the system is a prerequisite for system simulation. The CCGT plant consists of compressors, liquid pumps, turbines and valves besides a host of heat exchangers of various kinds. The strategy of system simulation is strongly dependent on the manner in which the characteristics of various components are available. For the purpose of system simulation, these characteristics are represented by information flow diagram, which is essentially a block diagram indicating that the output variables as known functions of the input variables. Often it is possible to rearrange the functional relationships, and therefore the choice of input and output variables to some extent are arbitrary. It is therefore possible (and necessary) to choose the input and output variables judiciously to arrive at an optimal simulation strategy. Modelling of the CCGT plant consists of three parts as follows,

1) Modelling of physical properties of the working fluids, here air, combustion gases and steam.

The air is modeled as a perfect gas with non-constant specific heat. The variation of specific heat at constant pressure Cp is normally modeled by several terms of a power series in temperature T. This expression is used in conjunction with the general thermodynamic equations to generate a gas table for particular gas. [10]

$$Cp = A_0 + A_1 T + A_2 T^2 + A_3 T^3 + A_4 T^4 + A_5 T^5 + A_6 T^6 + A_7 T^7 \quad (1)$$

The above equations are valid over a temperature range of 166 to 2225 Kelvin and fuel air ratio of 0 to 0.0676. By using this, the air and combustion products properties functions are developed in the air property program module.

Water and steam properties are modeled by the formulation released by the International Association for Properties of Water and Steam (IAPWS), [2].

2) Gas turbine and inlet air conditioner modelling, i.e., gas cycle modelling.

The inputs to the air conditioner are the ambient air and injected water parameters. By energy balance per unit mass of dry air, following equation is obtained.

$$(h_{air} + w h_w)_0 + (m_{water\,inj} h_{water\,inj})_0 =$$
$$(h_{air} + w h_w)_1 + (m_{water\,inj\,0} - (w_1 - w_0)) h_{water\,inj\,1} \qquad (2)$$

Equation 2 is an implicit function of the outlet temperature of the air conditioner, which is solved by iteration.

Across a compressor stage the temperature rise is:

$$T_2 / T_1 = (P_2 / P_1)^{(\gamma-1) / (\gamma * \eta_c)} \qquad (3)$$

And corresponding work done is given by:

$$W_{stage} = h_{2\,air} - h_{1\,air} + (h_{2\,water} - h_{1\,water}) \, m_{1\,water} \qquad (4)$$

The above equation holds true if the physical properties of working fluids are constant. But actually they vary with temperature and pressure. To account for this we assume that the total pressure rise is occurring through a large number of stages. The pressure and temperature rise across a stage is very small and hence the properties of working fluids can be assumed to be constant. In this way, the above equations are evaluated across all stages and the summation of work across all stages gives the total compressor work. In an over-spray condition, un-evaporated water will be present inside the compressor as the inlet air is fully saturated. As air is compressed adiabatically the temperature of air increases, thus bringing down the saturation, which promotes further evaporation of water between stages. For modeling this, after compression of air at outlet of each stage, the un-evaporated water at inlet of stage is being mixed with the compressed air adiabatically. Similar to inlet air conditioner, the enthalpy balance can be written as below:

$$H_{2\,air} + M_{1\,water} \, H_{2\,water} + M_{1\_water\_unevap} \, H_{1\_water\_unevap}$$
$$= H_{x\,air} + M_{x\,water} \, H_{x\,water} + \qquad (5)$$
$$(M_{1\,water\,unevap} - (M_{x\,water} - M_{1\,water})) \, H_{x\_water\_unevap}$$

Where $H_{x\,air}$, $H_{x\,water}$ are enthalpy of air and evaporated water after adiabatic mixing at the outlet of stage.

By mass balance, the amount of un-evaporated water at outlet after adiabatic saturation can be found as given below:

$$W_{unevaporated} = M_{1\_water\_unevap} - (M_{x\,water} - M_{1\,water}) \qquad (6)$$

Now the right hand side of energy balance equation, 5 is an implicit function of outlet temperature of the stage. This temperature is solved by bisection iteration in the program module. This gives the inlet temperature for next stage and this continues until all water is evaporated.

Input to combustion chamber includes the pressure loss, combustion efficiency; heat loss etc. The liquid fuel naphtha is modeled as $(CH_2)_n$. Two options are provided for the user in the combustor module, namely 1) specified fuel mass flow rate,

and 2) specified firing temperature. If one is specified, then the other can be calculated.

Applying energy balance across the combustor

$$h_{air2} + w \, h_{water\,2} + FA * LCV * \eta_{comb} = h_{air3} + w \, h_{water\,3} \qquad (7)$$

Above equation is iteratively solved for fuel air ratio, FA or Turbine Inlet Temperature, TIT as per the specified condition.

Across a turbine stage the temperature drop is:

$$T_4 / T_3 = (P_4 / P_3)^{(\gamma-1) / (\gamma * \eta_t)} \qquad (8)$$

Corresponding work done is given by

$$W_{stage} = h_{3\,air} - h_{4\,air} + (h_{3\,water} - h_{4\,water}) \, m_{3\,water} \qquad (9)$$

The above equation will hold true only if the physical properties of working fluids are constant. However, they actually differ depending upon the temperature and pressure. Assuming that the drop in pressure occurs in several stages, the properties of working fluids remain constant since the drop in temperature and pressure in a given stage is minimal. In this way, the above equations are evaluated across all stages and the summation of work across all stages gives the total turbine work. The blade cooling bleed loss is modeled by bypassing the specified cooling flow across the turbine first stage.

3) Heat Recovery steam generator, HRSG and Steam turbine modelling, i.e., steam cycle modelling.

The HRSG consist of number of heat exchangers arranged in series. These heat exchangers are basically cross flow of different configurations provided at various sections of the boiler to raise water temperature and superheat the steam before entry into the HP and LP turbines. The modeling of these heat exchangers involves determination of the pressure and temperatures of outgoing streams for given pressures, temperatures and flow rates of incoming streams. This is most conveniently done using the effectiveness concept.

$$Q = \in C_{min} (T_{hot} - T_{cold}) \qquad (10)$$

The effectiveness of these cross flow exchangers requires the calculation of heat transfer coefficients, which are available in standard heat transfer literature. In addition, the pressure drops in heat exchangers are calculated from standard correlations. The performance characteristics of various pumps used are taken from their characteristics curves.

The system simulation strategy is obtained by suitably combining the information flow diagrams of individual components of the system. Due to the non-linear nature of equations modeling the components, an iterative solution is required. This necessitates assumption of suitable initial values to start the simulation and check on the corresponding compatibility equations. Consequently, the whole task of HRSG simulation reduces to that of obtaining appropriate values of variables so that the corresponding compatibility equations are satisfied. These variables are then solved by fixed-point iteration. The solution gives the steam mass flows, flue gas and steams temperatures at various locations of HRSG

Finally the steam turbine work is calculated by the equation:

$$W_{turbine} = \eta_{turbine}(h_{2isen} - h_1) \qquad (11)$$

The efficiency and pressure ratio are taken from the characteristic curves of steam turbine. Steam mass flow and steam parameters come from output of HRSG module.

VALIDATION OF THE MODEL

The developed model of the combined cycle power plant is checked by comparing the model output and designed rated performance values provided by the original equipment manufacturer. Refer Table 1

Table 1 Comparison of predicted and design values

FUEL FLOW	MWGT		MWCC		MWST	
kg/s	RATED	COMPUTED	RATED	COMPUTED	RATED	COMPUTED
8.3	115.2	115.5	359.57	356.6	129.17	125.3
8.95	124.8	124.84	391.1	386.75	141.5	139.3
deviation %						
8.3	0.2604		-0.827		-3.0013	
8.95	0.0320		-1.112		-1.554	
FUEL FLOW	COMP AIR FLOW		HRSG EXIT TEMP		TOTAL STEAM FLOW	
kg/s	RATED	COMPUTED	RATED	COMPUTED	RATED	COMPUTED
8.3	394.17	394.3	117.9	120.4	126.61	125.302
8.95	395.83	396.1	117.1	119.6	139.17	135.82
deviation %						
8.3	0.0329		2.12041		-1.0330	
8.95	0.0674		2.134		-2.4071	
FUEL FLOW	GT EFFICENCY		CC EFFICENCY		GT EXHAUST TEMP	
kg/s	RATED	COMPUTED	RATED	COMPUTED	RATED	COMPUTED
8.3	31.6	31.5	49.3	48.6	553	553
8.95	31.63	31.64	49.56	49.01	588	581.5
deviation %						
8.3	-0.3164		-1.4198		0	
8.95	0.0316		-1.1099		-1.1054	

Legend : GT- Gas Turbine, CC- Combined Cycle
ST- Steam Turbine

As evident from the data above, the computed and rated values are excellently matching. Maximum deviation is about 3%.After developing the model for the CCGT plant, the effect of inlet air conditioner, i.e., swirl flash over spray system was analyzed to assess the potential benefits that can be obtained.

PARAMETRIC STUDY

With the help of developed model, analysis was carried out to compare the output and efficiency variation among the three methods viz. evaporative cooling, refrigerated inlet air cooling and swirl flash technique with respect to the rated conditions of 28 °C and a humidity of 60%. The results of the analysis are shown in Table 2. The comparative study reveals that the swirl flash technology is superior to the other methods. There is a maximum of 7.40% increase in the output and 2.37% in efficiency from the rated conditions at 2% overspray. The inlet air cooling is limited by the relative humidity in the case of evaporative cooling. Hence a maximum of only 1.95% increase in output and 0.44% increase in efficiency was possible with

**Table 2** Comparison of Gas turbine augmentation technologies

Method	Swirl Flash		Refrigerated cooling	Evaporative cooling
Condition	1%over spray	2%over spray	6°C	WBT - 22°C
Out put Change	2.61%	7.40%	6.87%	1.95%
Efficiency change	2.97%	2.37%	1.77%	0.44%

evaporative cooling. Refrigeration cooling is more effective, but can be reached only up to 5°C due to the risk of ice formation. There was an increase of 6.87% in output and 1.77% in efficiency possible with this technique.

The Swirl Flash Technology advantage

The analysis revealed the superiority of swirl flash technology over other methods in power and efficiency augmentation. Hence an in-depth analysis to learn the variation of power and efficiency improvements with respect to the possible changes in the ambient conditions of the locality was mandatory. In order to understand comprehensively the effect of swirl flash over spray, a sensitivity study by varying the ambient temperature and humidity ratio was conducted on the RGCCPP-NTPC, plant model. Program for modelled plant was run at different conditions like base case, 1% over spray, and 2% over spray to assess the performance of the CCGT plant to various operating conditions. Five values of ambient temperature (18°C, 23°C, 28°C, 33°C and 38°C) and six humidity ratio (50%, 60%, 70%, 80%, 90% and 100%) were considered for study. This is the maximum variation normally experienced in the site. Thus, a total of 90 cases were analyzed for the CCGT plant. The results are summarized with charts. Data is presented, as a change expressed in percent, with respect to the corresponding value at base case, no over spray, 28 °C and 60% humidity i.e., the rated conditions.
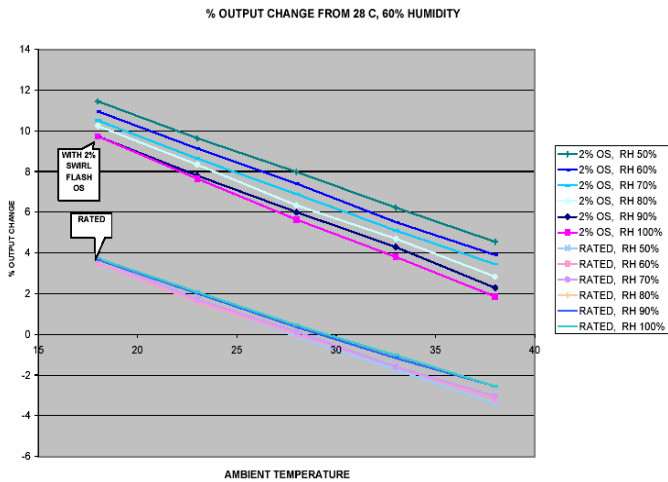
Figure 5 Power out put change at rated conditions
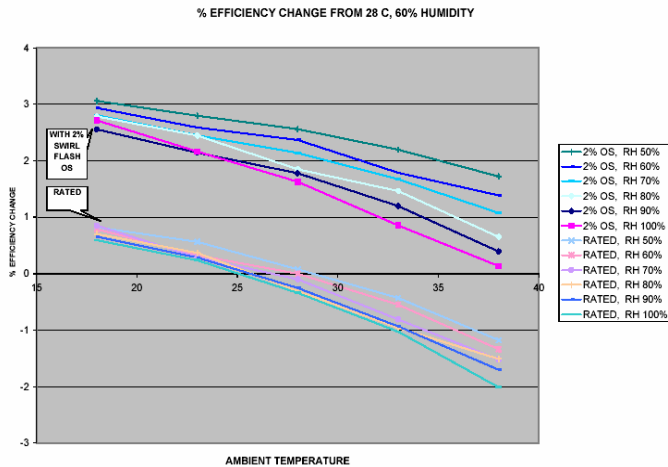
Figure 6 Efficiency change at rated conditions

Comparison of the output and efficiency variation at rated conditions and with 2% overspray are depicted in figures 5 and 6. The bottom six lines in figures 5 and 6 shows the percentage combined cycle output and efficiency drop with ambient temperature at various humidity conditions without any swirl flash spray. The top six lines in figures 5 and 6 shows the percentage combined cycle output and efficiency drop with increase in ambient temperature at various humidity conditions with 2% swirl flash overspray. With overspray, the output and efficiency changes significantly with humidity at same ambient temperature. For example at 28° Celsius with 2% overspray the output gain is 8% at 50% humidity compared to 5.8% gain at 100% humidity. This is due to the evaporative cooling happening at inlet of compressor at low humidity conditions.

The figures 5 and 6 clearly shows the superior benefit offered by swirl flash overspray system. It can be seen that even at the worst ambient condition of 38° Celsius swirl flash with 2% overspray gives a net gain of 3.9% in output and 1.39% in efficiency. The summary of the results are presented

in Table 3 and Table 4. It can be seen that with 2% over spray, the percentage output gain at rated condition is about 7.4% and the percentage efficiency gain at rated condition is 2.37%.

Table 3 Summary of findings (out put change)

CONDITION	CURRENT	1% OS	2% OS
BEST 18°C	3.47%	6.42%	10.96%
RATED 28C	0%	2.61%	7.4%
WORST 38C	-3.22%	-0.47%	3.9%

Table 4 Summary of findings (efficiency change)

CONDITION	CURRENT	1% OS	2% OS
BEST 18C	0.79%	3.85%	2.94%
RATED 28C	0%	2.97%	2.37%
WORST 38°C	-1.34%	2.13%	1.39%

CONCLUSIONS

From the simulation analysis it was established that out of the three techniques, the Swirl Flash is a superior water injection technology for gas turbine power and efficiency augmentation.

The model study showed an increase in power out put of 7.4% at rated conditions and efficiency gain of 2.37% for the combined cycle plant under study at 2% overspray. With the restrictions on humidity and temperature, evaporative cooling offered only a maximum enhancement of 1.95% in power and 0.44% in efficiency. The values are 6.87% and 1.77%.respectively for refrigerate cooling.

Due to the superior NOx reduction achieved, the swirl flash system is more environment friendly. Combinations of swirl flash technology with evaporative cooling or refrigerate cooling can yield higher enhancements in power and efficiency. A preliminary economic analysis revealed that additional investment on power enhancement can be realized within a payback period of just 6 months.

NOMENCLATURE

$A_0 - A_7$	[-]	*Constants*
C_p	[KJ/kgK]	Specific heat at constant pressure
C_{min}	[KJ/K]	Minimum heat capacity
h	[KJ/kg]	Enthalpy
m	[kg]	Mass
P	[bar]	Pressure
Q	[W]	Heat transfer rate
T	[K]	Temperature
w	[kg/kg dry air]	Humidity ratio

Special characters

η	[-]	Efficiency
γ	[-]	Ratio of specific heats

Subscripts

0,1,2,3,4	Appropriate stages
c	Compressor
comb	Combined
min	Minimum
t	Turbine

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