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ANALYSIS OF EVAPORATIVE AND VAPOUR COMPRESSION SYSTEMS USED FOR COOLING OF COMPRESSOR INTAKE AIR OF GAS TURBINE PLANTS

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ABSTRACT
Compressor intake air cooling in Gas turbine power plants is considered to be very important and feasible requirement now-a-days. The methods for the compressor intake air cooling are evaporative cooling, fogging, vapour compression, absorption chiller, hybrid system utilizing absorption chiller and vapour compression system and hybrid system utilizing evaporative cooling and absorption chiller system. Evaporative cooling and fogging are the most simple and economical as compared to other methods, to implement in the power plants. Hybrid systems give the additional advantage of coupling the benefits of two individual cooling methods. This system can decrease the chilled water temperature upto approximately 3.5ºC. In the present work, the focus is upon the performance analysis of two cooling methods- evaporative cooling and vapour compression system. Evaporative cooling is the simplest one and results in air temperature reduction of 13.5, 14.9 and 2.6ºC in moderate, hot and dry, and cold and humid ambient air conditions respectively. Vapour compression system is highly efficient but uses high quality energy as the system input, thus, is less economic.

Keywords:
Gas Turbine; Evaporative cooling system; Vapour compression cooling system

1. INTRODUCTION

The adverse effect of high ambient air temperatures on the power output of a gas turbine is two fold: as the temperature of the air increases the air density decreases and consequently the air mass flow The reduced mass flow directly causes decrease in the power output of the gas turbine .On the other hand the higher intake air temperature results in an increase in specific compressor work. Thus the use of high temperature ambient air results in a net decrease in the gas turbine output. The most common approach utilized in power generation to increase mass flow is to increase the air density by lowering the inlet air temperature. Depending on the type of the gas turbine, the electric output will decrease by-a percentage between 6 and more than 10 % for every 10° C of intake air temperature increase. At the same time, the specific heat consumption increases by a percentage between 1.5 % and more than 4.5 %. It can be concluded that at temperature of 40-45°C, common in India and various other countries where a large number of gas turbines are used for electricity generation, there is a power loss of more than 20%, combined with a significant increase in specific fuel consumption, compared to ISO standard condition (15°C).Thus in summer over a long period of time, gas turbines demonstrate a lower power output and efficiency than the equipment could actually perform. If it was possible to obtain a constant low inlet air temperature, a constant high power output could be generated from a gas turbine.

1.1 Evaporative Cooling System

Evaporative cooling is based on the evaporation of water injected in the intake air of the gas turbine. As water evaporates, the latent heat of evaporation is absorbed from the water body and the surrounding air. As a result, both the water and the air are cooled during the process. In the limiting case, the air leaves the cooler at a saturated state.. Therefore, the evaporative cooling process follows a line of constant wet-bulb temperature on the psychrometric chart. The gain from the use of an evaporative cooler depends upon the relative humidity of the ambient air and it is high for dry ambient conditions, whereas the gain for wet ambient conditions is low. This indicates that this type of intake air-cooling could mainly be of interest in countries where the climate is hot and dry. Furthermore, these coolers are limited by the amount of moisture in the air. Once saturation is reached, evaporative cooling systems are unable to evaporate more water into the air stream. For this reason, in hot and humid regions, it is not often possible to accomplish more than about 5.5 to 8.5°C of cooling.

1.2 Vapour Compression (V-C) System

Vapour Compression System is shown in Figure 1.1. This system comprises mainly of the following components: evaporator, compressor, condenser and throttle valve. Figure 1.7 shows the temperature (T) versus entropy (s) diagram for the system. The evaporation of the refrigerant inside the evaporator takes place at a constant

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temperature and pressure (process a-b). Evaporation process is followed by a compression process (process b-c') which in ideal case is assumed to be isentropic. The condensation of high pressure and temperature refrigerant vapour leaving compressor takes place inside condenser at a constant pressure (process c'-d).

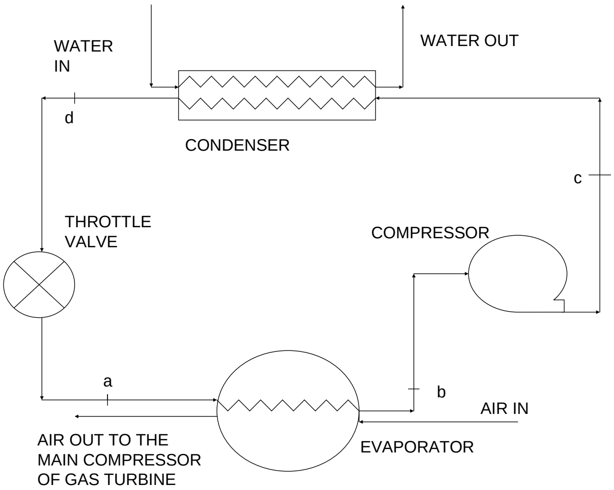

Figure 1.1 Vapour compression system

2. LITERATURE REVIEW

Compressor intake air cooling is developing to be an important area of research in the Modern Gas Turbine Power Plants. Kumar *et.al.* [1], investigated the improved gas turbine efficiency using spray coolers and through Alternative Regeneration Configuration. Gomez *et. al.* [2], presented the manufacture, test bed setup and trials carried out on a ceramic evaporative cooling system which acts as a semi-indirect cooler. The tests presented, show the system behavior for various supply air conditions. Datta *et. al.* [3], fabricated and tested 8.5 ton indirect-direct evaporative cooling system and the performance of the system was compared with a computer prediction. The system's scope for use in India and Australia was analyzed. Ondryas *et. al.* [4], investigated the gas turbine power augmentation in a cogeneration plant using inlet air chilling. Kakaras *et. al.* [5], simulated the results for two test cases: a simple cycle gas turbine and a combined cycle plant. Al-Amiri *et. al.* [6], assessed the benefits of incorporating combustion turbine inlet air cooling systems into a reference combustion turbine plant, which was based on a simple cycle under base load mode. Alhazmy *et. al.* [8], studied the performance enhancement of gas turbine power plants using spray cooling (water spraying system and cooling coil). Spray cooler reduces the temperature of incoming air by 3-15°C enhancing the power by 1-7% and improving efficiency by 3%. Cammarata et. al. [10], investigated an application of exergonomic theory to an air-conditioning system for optimization purposes.

3. FORMULATION

3.1 Evaporative Cooling System

The effectiveness (ε) of the system is defined as the ratio of the difference in dry bulb temperature across the cooler to the difference in the inlet air dry bulb temperature and wet bulb temperature. Accordingly,

$$\varepsilon = (T_0-T_1)/(T_0-T_{wb}) \quad (3.1)$$

or, $$T_1= T_0-(T_0-T_{wb})\,\varepsilon \quad (3.2)$$

where,
T_0= Ambient inlet air dry bulb temperature in K.
T_1= Dry bulb temperature of exit air in K.
T_{wb}= Wet bulb temperature of the ambient inlet air in K.
Reduction in temperature of air (ΔT) is given by

$$\Delta T= T_0 - T_1 \quad (3.3)$$

Mass flow-rate of inlet air is given by

$$M_{a,\,inlet}=V_1/\,v_0 \quad (3.4)$$

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Mass flow-rate of exit air given by

$M_{a,\ exit} = (V_1 / v_1)$ (3.5)

and increase in mass flow-rate of air is given by

$\Delta M = (V_1 / v_1) - (V_1 / v_0)$ (3.6)

where,

V_1= Volumetric air flow-rate in cu.m/s.

v_1= Specific volume of exit air at temperature T_1 in cu.m/kg.

v_0= Specific volume of inlet air at temperature T_0 in cu.m/kg.

$M_{a,\ inlet}$= Mass flow-rate of inlet air in kg/s.

$M_{a,\ exit}$ = Mass flow-rate of exit air in kg/s.

ΔM= Increase in mass flow-rate of air in kg/s

The inlet air specific humidity is given by

$\omega_0 = 0.622\ P_{v0} / (P_{atm} - P_{v0})$ (3.7)

where,

ω_0= Specific humidity of ambient inlet air in kg/kg of dry air.

P_{atm} = Atmospheric pressure (101.325 kPa).

P_{v0}= Partial pressure of water vapour at the inlet air in kPa.

The exit air specific humidity (ω_1) is obtained by using the energy balance equation for the cooler section [Dawoud *et. al* (8)] and is given by

$\omega_1 = [(C_{pa0}T_0 - C_{pa1}T_1) + \omega_0 (C_{pv0}T_0 + h_{fg0} - C_{ps}T_1)] / [C_{pv1}T_1 + h_{fg0} - C_{ps}T_1]$ (3.8)

$\varphi_0 = (\omega_0/0.622)\ [(P_{atm} - P_{v0})/ (P_{s0})]$ (3.9)

φ_0= Relative humidity of inlet air in kg/kg of water vapour.

P_{s0}= Saturation pressure of water at inlet air temperature (T_0) in kPa.

Exit air relative humidity is given by:

$\varphi_1 = (P_{atm}\ \omega_1/ (0.622 + \omega_1))/ P_{s1}$ (3.10)

φ_1= Exit air relative humidity in kg/ kg of water vapour.

P_{s1}= Saturation pressure of water vapour at the exit air temperature (T_1) in kPa.

$Q_e = (V_1 / v_0)\ C_{pa}\ (T_0 - T_1)$ (3.11)

The formula for make-up water requirement (m_w) is as follows:

$m_w = (V_1 / v_0)\ (\omega_1 - \omega_0)$ (3.12)

m_w= Mass flow-rate of make-up water in kg/s.

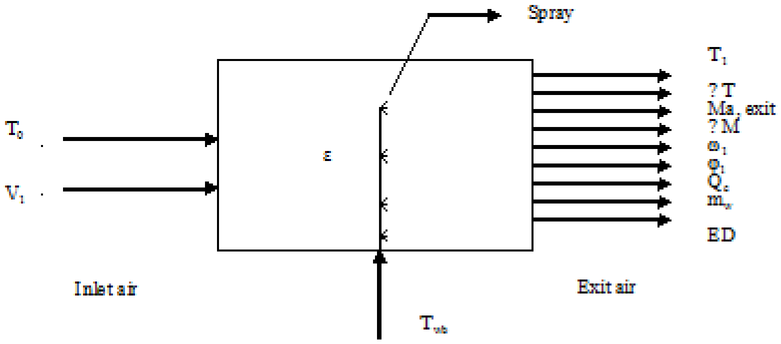

Figure3.1 Numerical model for evaporative spray cooler

As shown in Figure 3.2, the various input parameters in the model are: inlet air temperature (T_0), volumetric air flow-rate (V_1), effectiveness of the system (ε) and the wet bulb temperature of inlet air (T_{wb}). The output parameters are: exit air temperature (T_1), reduction in temperature of air (ΔT), mass flow-rate of exit air ($M_{a,\ exit}$), increase in mass flow-rate of air (ΔM), specific humidity of exit air (ω_1), relative humidity of exit air (φ_1), cooling load (Q_e), mass flow-rate of make-up water (m_w) and exergy destruction in the system (ED), obtained from equations (3.2) to (3.15) respectively.

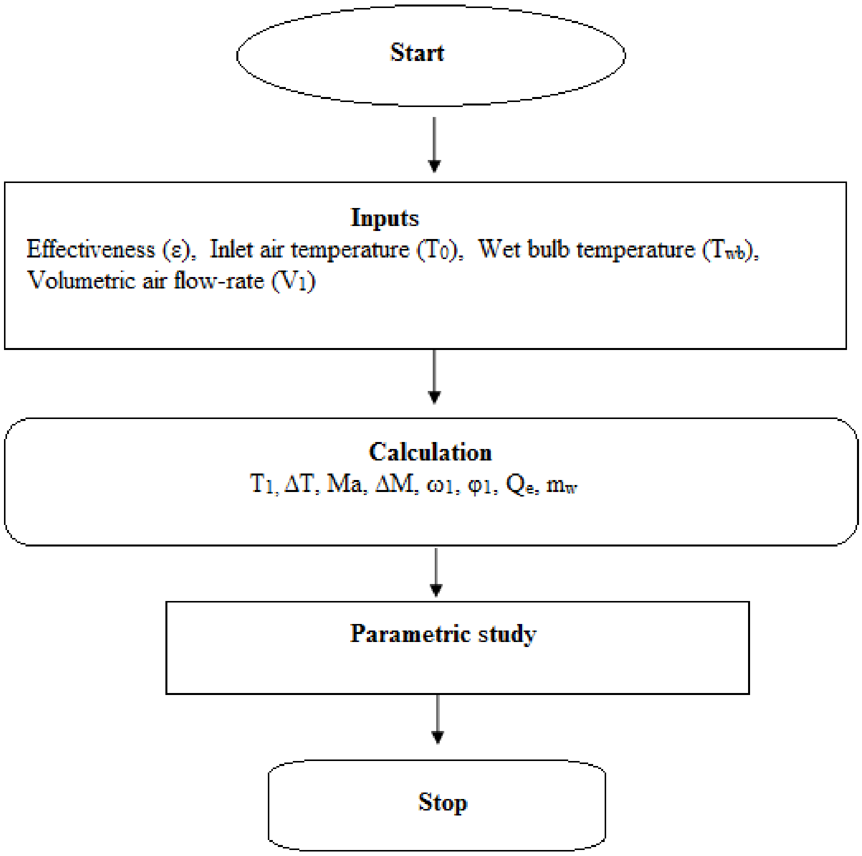

Figure 3.2 Flow chart for calculation of evaporative cooling system

Evaporative cooling system calculation

Analysis of evaporative cooling system is done for three different inlet air conditions. The outlet conditions are calculated for spray effectiveness varying from 0.8 to 1.0. The results are shown in Tables 3.1 and 3.2 The values of volumetric flow-rate (V_1) and atmospheric pressure used in the calculation are 86 cu.m/s and 101.325 kPa respectively.

Table 3.1 Output of evaporative cooling system for moderate ambient condition, [T_0 =308.15 K, T_{wb}=294.56 K, $M_{a,\ inlet}$= 98.4962 kg/s, (Equivalent inlet air conditions: ω_0=0.011 kg/kg, φ_0= 30 percent)]

E	T_1 (K)	ΔT (K)	$M_{a,\ exit}$ (kg/s)	ΔM (kg/s)	ω_1(kg/kg)	φ_1(in percent)	Q_e (kW)	M_w (kg/s)
0.8	297.2200	10.872	99.3351	2.5082	0.0168	80.00	1060.3593	0.6339
0.9	295.9190	12.231	99.6655	2.8381	0.0176	90.52	1192.7476	0.7119
1.0	294.5600	13.590	100.0	3.1719	0.0184	100.00	1325.1010	0.7897

Table 3.2 Output of evaporative cooling system for hot and dry ambient condition, [T_0= 323.15 K, T_{wb}=308.15 K, $M_{a,\ inlet}$=93.9242 kg/s, (Equivalent conditions of air: ω_0=0.0318 kg/kg, φ_0= 40 percent)]

E	T_1	ΔT	$M_{a,\ exit}$	ΔM	ω_1(kg/kg)	φ_1(in	Q_e	M_w

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	(K)	(K)	(kg/s)	(kg/s)		percent)	(kW)	(kg/s)
0.8	311.15	12.0	91.9178	2.3693	0.0376	80.0717	1085.2832	0.6857
0.9	309.65	13.5	92.2316	2.6826	0.0385	88.7330	1220.7679	0.7699
1.0	308.15	15.0	92.5492	2.9944	0.0395	98.3485	1356.2135	0.8537

From the results shown in Tables 3.1 and 3.2, it can be concluded that exit air temperature (T_1), reduction in air temperature (ΔT), mass flow rate of exit air ($M_{a,\ exit}$), increase in mass flow rate of air (ΔM), specific humidity of exit air (ω_1), relative humidity of exit air (φ_1), cooling load (Q_e) and mass flow-rate of make-up water increase with increase of effectiveness (ε) of evaporative cooler.

4. Vapour Compression System

Cooling load:

$$Q_e = (P_1 V_1 / RT_1)\, C_{pa}\, (T_0 - T_1) \qquad (4.1)$$

where,

P_1= Inlet pressure to the main compressor of the gas turbine cycle in kPa

R= Characteristic gas constant of air (0.2871 kJ/kg K)

V_1 = Volumetric air flow rate in cu.m/s

C_{pa}= Average specific heat of air in kJ/kgK

T_0= Ambient inlet air dry bulb temperature in K

T_1= Dry bulb temperature of exit air in K

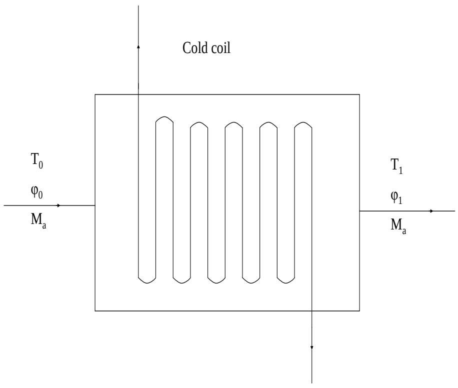

Figure 4.1 Cooling coil

Refrigeration power:

$$W_r = (P_1 V_1 / R)\,(C_{pa} / \eta_r)\,(T_0/T_1 - 1)\,[(T_{hc}\,(\exp(XT_1) - 1) / T_1 \exp(XT_1) - T_0) - 1)] \qquad (4.2)$$

η_r= Refrigeration cycle isentropic efficiency

T_{hc}= Refrigeration higher isotherm (condenser temperature) in K

And

$$X = RUA_e / C_{pa}\, V_1 P_1$$

. UA_e= Evaporator heat conductance in kW/K

.

Ratio of mass flow-rate of air at exit air temperature (T_1) to that at inlet air temperature (T_0) per unit volumetric air flow-rate can be expressed as

$$m_{r1} = (M_a)_{exit} / (M_a)_{inlet} \qquad (4.3)$$

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$$= T_0 / T_1 \quad (4.4)$$

Calculation

Based upon the formulation given in Section 4.1, results of vapour compression system for unit volumetric air flow-rate and variable volumetric air flow-rate are shown in Table 4.1 and table 4.2 respectively.

Table 4.1 Output of vapour compression system for unit volumetric air flow-rate

T_1 (K)	Q_e (kW)	m_{r1}	W_r (kW)	Q_c (kW)
299.0000	1.1900	1.0033	0.4845	1.6744
295.0000	6.0281	1.0169	2.6378	8.6659
290.0000	12.2581	1.0345	5.8499	18.1080

Input parameters: T_0=300 K, P_1=101.325 kPa, V_1=1 cu.m/s, η_r=0.6, UA_e=2.98 kW/K, T_{hc}=372.9 K

Table 4.2 Output of vapour compression system for variable volumetric air flow-rate

ΔT (K)	T_1 (K)	Q_e (kW)	φ_1 (in %)	W_r (kW)
8.0	300.15	948.8350	56.4997	735.4411
10.0	298.15	1193.7693	63.5973	649.3759
12.0	296.15	1441.9189	71.7167	564.4497

Input parameters: T_0=308.15 K, T_{wb}=296.15 K, P_1=101.325 kPa, V_1= 100 cu.m/s,T_{0ref}=298.15 K,

From Table 4.2, it can be concluded that refrigeration power increases with decrease in exit air temperature. In case of gas turbine cycle, this refrigeration power will be supplied through gas turbine power output.

5.3 Validation of program

The calculated values of the system parameters by the program are compared with the sample results of Ahmadul Ameen [15]. The input parameters are listed below for the tabulated results obtained (refer Table 5.1)

Table 5.1 Comparision of obtained data with Ahmadul Ameen [12]

	Q_e (kW)	Q_2 (kW)	Q_c (kW)
Present formulation	383.7605	458.8300	409.0707
Ahmadul Ameen [28]	383.7600	458.7800	409.0700
Percentage deviation (%)	0.0001	0.0109	0.0002

Input parameter: P_1=101.325 kPa, T_0=300.15 K, T_1=283.15 K, m_1=0.55 kg/s, T_2= 323.15 K, T_{hc}=309.15 K, T_a=301.15 K, T_g=373.15 K, T_{ce}=283.15 K.

6. CONCLUSIONS

Evaporative cooling offers degree of cooling varying between 2.72 to 15 °C for the three ambient air conditions common in India i.e., moderate, hot and dry and cold and humid This system is a simple one and easy to operate and involves lower costs compared to other systems. But the system has limitation that it can not be used to cool air below wet bulb temperature of air.

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Vapour compression system is a highly efficient system but the running cost and maintenance of the system is very high because it involves too many rotating components. As shown in the results in Chapter 4, for unit volumetric air flow-rate, the cooled air temperature varies between 285 to 299 K and the refrigeration power and refrigeration load varies between 9.70 to 0.48 kW and 18.70 to 1.19 kW respectively. For variable volumetric air flow-rate, the above parameters vary between 316.27 to 735.44 kW and 2206.32 to 948.83 kW respectively.

It can be concluded that choice of a particular depends upon the desired characteristics and location of the power plant. Evaporative cooling system and absorption chiller system are the best choice for the economy and efficiency purpose.

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