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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. Hybrid system utilizing vapour absorption chiller and vapour compression system is a series type system using two 50 percent capacity trains of absorption and mechanical chillers. This system can decrease the chilled water temperature upto approximately 3.5ºC. On the other hand hybrid system utilizing evaporative cooling and absorption chiiling system is largely flexible and can cool the inlet air upto 5 ºC independent of ambient air temperature and humidity as compared to the conventional systems which cannot go much below 10 ºC. 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. 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 9.7 to 0.5 kW and 18.7 to 1.2 kW respectively.
Keywords: Gas Turbine; Evaporative cooling system; Vapour compression cooling system

Nomenclature

ω_0	Specific humidity of ambient inlet air in kg/kg of dry air.
ω_1	Specific humidity of exit air
φ_0	Relative humidity of inlet air in kg/kg of water vapour.
φ_1	Exit air relative humidity in kg/ kg of water vapour
ε	Effectiveness of evaporative cooler
η_r	Refrigeration cycle isentropic efficiency
C_{pa}	Average specific heat of air in kJ/kgK
m_w	Mass flow-rate of make-up water in kg/s
$M_{a, inlet}$	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
P_1	Inlet pressure to the main compressor of the gas turbine cycle in kPa
P_{atm}	Atmospheric pressure (101.325 kPa).
P_{s0}	Saturation pressure of water at inlet air temperature (T_0) in kPa.
P_{s1}	Saturation pressure of water vapour at the exit air temperature (T_1) in kPa.
P_{v0}	Partial pressure of water vapour at the inlet air in kPa.
Q_e	Cooling load (kW)
Q_c	The heat loss in condenser is given by
R	Characteristic gas constant of air (0.2871 kJ/kg K)
T_0	Ambient inlet air dry bulb temperature in K.
T_1	Dry bulb temperature of exit air in K.
T_{hc}	Refrigeration higher isotherm (condenser temperature) in K
T_{wb}	Wet bulb temperature of the ambient inlet air in K.
ΔT	Reduction in temperature of air is given by

UA_e Evaporator heat conductance in kW/K
V_1 Volumetric air flow-rate in cu.m/s.v_1= Specific volume of exit air at temperature in cu.m/kg
v_0 Specific volume of inlet air at temperature T_0 in cu.m/kg.
X Parameter represents the ratio of heat conductance of refrigeration cycle evaporator

I. 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.

A. 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.

B. 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 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).

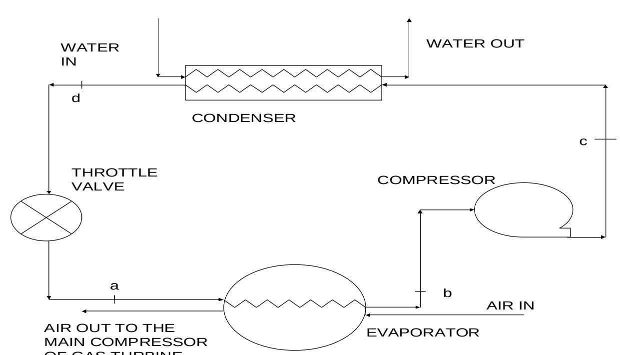

Figure1.1: Vapour compression system

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II. 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. Dawoud et. *al.* [7], evaluated the power requirements of several inlet air cooling techniques for gas turbine power plants in two locations: Marmul and Fahud in Oman. Fogging cooling was accompanied with 11.4% more electrical energy in comparision with evaporative cooling in both locations. The LiBr-H_2O cooling offered 40% and 55% more energy than fogging cooling at Fahud and Marmul, respectively. 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%. Camargo *et .al.* [10], presented the mathematical model of the evaporative cooling process for human thermal comfort. The results of experimental tests in a direct evaporative cooler in Air Conditioning Laboratory at the University of Taubate Mechanical Engineering Department were also compared with mathematical model. Bhargava *et. al.* [11], did a parametric analysis on the effects of inlet fogging and evaporative conditions on a wide range of existing gas turbines. Arora [12], presented a basic as well as applied thermodynamic treatment of refrigeration and air conditioning in a very comprehensive manner. A sound physical basis had been laid for obtaining fairly precise estimates of refrigeration and air-conditioning equipment.

III. FORMULATION

A. *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,

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

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

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

Mass flow-rate of inlet air is given by

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

Mass flow-rate of exit air given by

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

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

The inlet air specific humidity is given by

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

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

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

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

Exit air relative humidity is given by:

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

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

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

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

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

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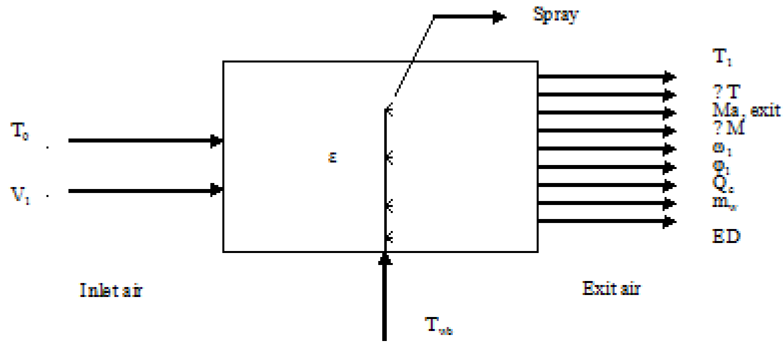

Figure3: Numerical model for evaporative spray cooler

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.

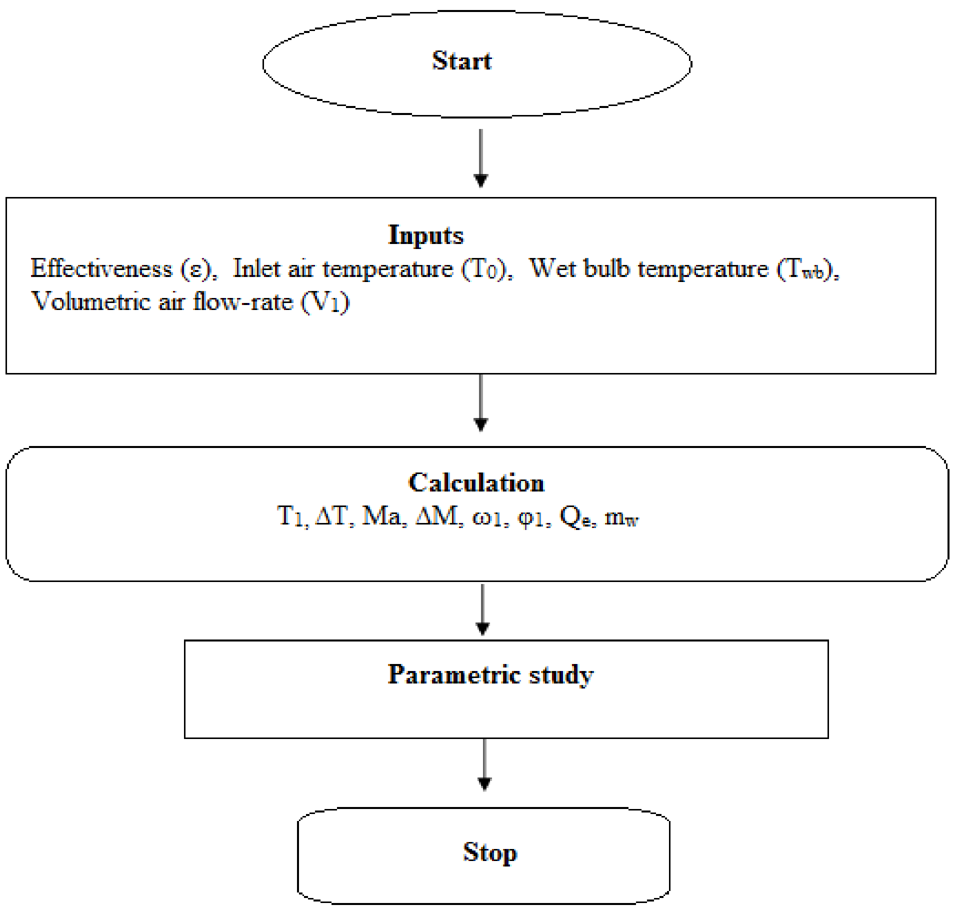

Figure 3.2: Flow chart for calculation of evaporative cooling system

B. 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.

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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 (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	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.

IV. VAPOUR COMPRESSION SYSTEM

A. *Cooling load*

$$Q_e = (P_1V_1/ RT_1)\, C_{pa}\, (T_0 - T_1) \qquad (4.1)$$

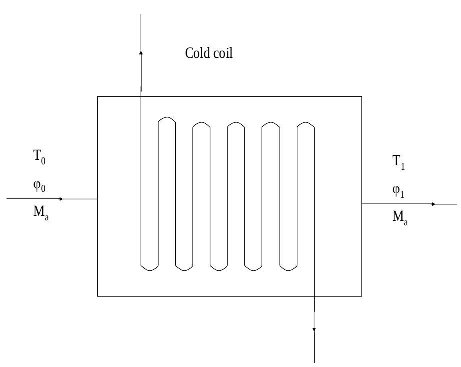

Figure 4.1: Cooling coil

Refrigeration power:

$$W_r = (P_1V_1/R)\, (C_{pa} / \eta_r)\, (T_0/T_1 -1)\, [(T_{hc}\, (\exp (XT_1) - 1)/ T_1 \exp (XT_1) - T_0)- 1)] \qquad (4.2)$$

$X = RUA_e/C_{pa}\, V_1P_1$

The parameter X represents the ratio of heat conductance of refrigeration cycle evaporator per degree of T_1 to the thermal capacitance of the inlet air being refrigerated.

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

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

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

The heat loss in condenser is given by

$$Q_c = W_r + Q_e \qquad (4.5)$$

B. 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

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

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
285.0000	18.7007	1.0526	9.7011	28.4018

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

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,

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
14.0	294,15	1693.3492	81.0227	480.6396

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.

V. VALIDATION OF PROGRAM

The calculated values of the system parameters by the program are compared with the sample results of Ahmadul Ameen [14]. 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 [14]

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.

	Q_e (kW)	Q_a (kW)	Q_g (kW)	Q_c (kW)	Q_{ex}(kW)
Present formulation	383.7605	458.8300	484.6510	409.0707	28.0495
Ahmadul Ameen [28]	383.7600	458.7800	484.6500	409.0700	28.0500
Percentage deviation (%)	0.0001	0.0109	0.0002	0.0002	0.0018

VI. CONCLUSIONS

Evaporative cooling offers degree of cooling varying between 2.72 to 15 ºC 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.

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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