A COMPUTER PROCEDURE FOR GAS-TURBINE POWER AUGMENTATION BY FOG-COOLING

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ABSTRACT

This paper describes a procedure for evaluating the gains from increased power and reduced heat rate of a gas turbine that result from pre-cooling the turbine's inlet-air by fogging. Unlike conventional media-type evaporative cooling, fogging can achieve 100% saturation of the air so that its temperature can be reduced to the wet-bulb temperature. The present procedure consists of two main sub-models; (i) an inlet-air evaporative cooling sub-model and (ii) a gas-turbine performance sub-model. The inlet-air cooling sub-model applies thermodynamic principles to determine the cooled air temperature prior to the compressor. The gas-turbine performance sub-model estimates, from the turbine's characteristic curves, the resulting improvements in the turbine's power and heat-rate and gives the expected hourly revenues. For the user's convenience, the procedure has been computer-programmed using MATLAB. The paper verifies the procedure against relevant published data before studying the effects of the main parameters on the fogging-system's performance.

Keywords: Gas turbine, power-augmentation, fog-cooling, evaporative cooling, heat rate.

1. INTRODUCTION

Gas turbines are versatile power-generation machines but their performance is greatly degraded by adverse ambient conditions such as high air temperatures and humid or dusty environments. Having a constant volume flow rate, the power of the gas turbine is directly proportional to the mass flow rate of the air passing through it, which is directly proportional to the air density. Since the air-density is reduced by a high ambient temperature, gas turbines designed to operate at a standard condition of 15.6^{o}C (60^{o}F) lose significant portions of their generating capacity when installed in hot climates. A high inlet-air temperature also increases the compressor's work and lowers the plant thermal efficiency [1-3]. Therefore, gas turbines operating under hot climates do not only produce less power than their design capacity, but also consume more fuel. According to McCracken **[2]**, gas turbines produce 25-35% less power in summer than in winter at 5-10% higher heat rate (i.e. an average increase of 6% in fuel consumption).

A significant portion of the thermal generation of the National Electricity Corporation (NEC) comes from gas-turbines located at Khartoum North area. These turbines are usually used to meet the electrical demand at the peak-hours. Since the peak-load normally happens during the hot midday hours -when the gas turbines are least productive- the turbines' deficiency contributes to the noticeable shortage of electricity during summer time which causes big losses to NEC and considerable waste of resources to the whole country. While in temperate climates this problem is faced during the hot summer days only, in Sudan the air temperature is relatively high all the year around. Solving the problem by adopting power-augmentation methods does not only optimise the use of national resources, but also reduces the environmental impact of power generation near Khartoum.

Many power-augmentation methods can be used to compensate for the effects of ambient conditions on the gas-turbine's output, but the two most common methods are those of cooling the inlet-air and injecting water or steam into the combustion chamber **[4-6]**. Water/steam injection has the advantage of reducing the plant's NOx emissions while increasing its generation capacity. However, the amount of water/steam that can be injected is limited by factors such as the flame stability in the combustion chamber and the restriction that the secondary flow of water/steam should not obstruct the main air flow through the system. Although water is more effective than steam in boosting the turbine's output and reducing NOx emissions, it has the disadvantage of increasing the heat rate **[4, 6].**

Inlet-air cooling typically increases the gas turbine's output by 10% - 18% for every 10°C of decrease in inlet air temperature **[2]**. Like steam injection, inlet-air cooling reduces the heat-rate of the plant. Compared to steam or water injection, inlet-air cooling has the advantage that it does not interfere much with the normal operation of the system because it is external to it. The inlet-air temperature can be reduced evaporatively by directly spraying water, liquefied air, or liquefied natural gas (LNG), into it. Air cooling can also be achieved indirectly by a cooling coil carrying a cold fluid (water or refrigerant). This method can be made more effective by using thermal energy storage (TES) systems to make use of the low-cost, off-peak electricity **[7-9]**. It can also be combined with dehumidification in the hot and humid coastal and tropical regions **[10].**

Fog cooling is an evaporative cooling method that is becoming increasingly popular for air-conditioning applications in general and gas-turbine power augmentation in particular **[11]**. A series of stainless steel-tubing arrays distribute demineralised water under high pressure (14 - 25 MPa) to specially designed nozzles which, in turn, atomise the water into fine droplets in the form of fog. Due to its small size (5-10 μm) and distribution over a large area, the water droplets evaporate quickly and effectively cool the air. While pressurising the liquid water requires a minimal amount of work input, it significantly improves the vaporisation and cooling processes. Unlike media-type evaporative cooling, which can only achieve about 90% saturation, fogging can achieve full saturation of the inlet air and can cool it down to the wet-bulb temperature.

While conventional media-type cooling is difficult to recommend for highly humid environments like those of tropical and coastal areas, the effectiveness of fog-cooling make it a competitive power augmentation method for gas turbines even in these regions. An additional advantage of the fog-cooling system is that it can be used to spray more water fog than just that needed to saturate the air. The water "over-spray" vaporises inside the compressor and acts like an inter-cooler. This "wet-compression" process reduces the compression work and further increases the turbine's output.

The effectiveness and economical feasibility of the fog-cooling system is governed by the ambient conditions, the installation and operation costs, the electricity tariff, and the load-variation curve. The atmospheric conditions that control the effectives of the system are mainly the ambient temperature and humidity. The operation costs include the maintenance cost, the cost of fuel, and the cost of demineralised water. Although the impact of inlet-air pre-cooling on the turbine power is usually greater than that on the heat-rate, a greater number of the turbine's annual operating hours make the benefit of the heat-rate improvement similarly desirable.

A procedure that estimates the effects of the different factors on the fogging system performance can be a valuable aid for estimating the hourly, daily or annual revenues that result from improving the power and heat rate of the gas turbine. This paper describes such a procedure and the computer program based on it. The computer procedure is verified against relevant published data before being used to perform a parametric study on the effectiveness of the fogging system.

2. THE COMPUTER PROCEDURE

The present analytical procedure consists of two main steps or sub-models; (a) an inlet-air evaporative cooling sub-model and (b) a gas-turbine performance sub-model. The inlet-air evaporative cooling sub-model applies thermodynamic principles to determine the inlet-air temperature after the fogging system. The gas-turbine performance sub-model estimates the resulting improvements in the turbine power and heat-rate from the turbine's characteristic curves and determines the expected revenues. The following sections describe the two sub-models in more details.

2.1. The Inlet-Air Evaporative Cooling Sub-model

Figure 1 shows a diagram of the fog-cooling system. Point 1 is the original ambient condition and point 3 represents air after it undergoes cooling. Pressurised liquid water is sprayed into the air at point 2. The temperature of air after fog cooling can be obtained from an energy balance on the dry air, water spray, and air-borne water vapour before and after the system.

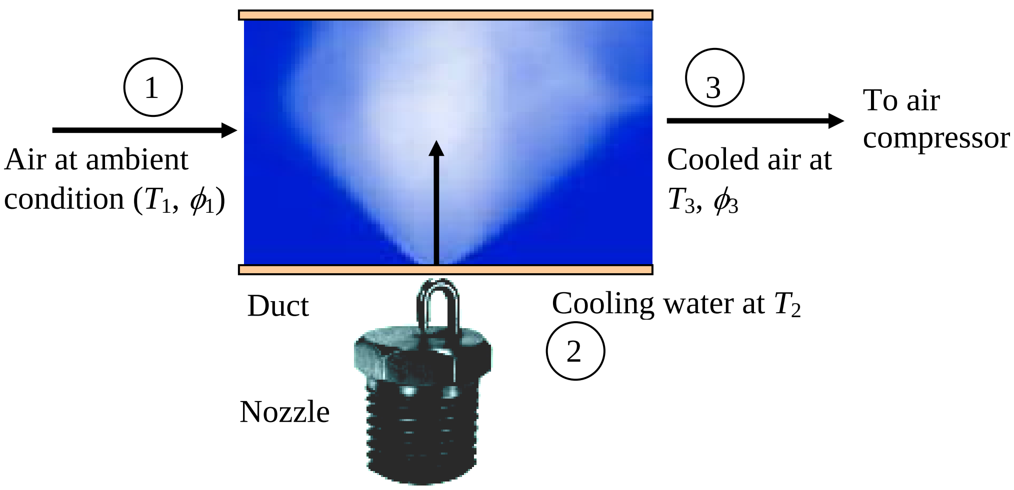


FIGURE 1: The fog-cooling system

Assuming adiabatic mixing, the energy gained by the sprayed water is balanced by the energy lost by the atmospheric air, after cooling so that:

$$m_w (h_{v3} - h_{w2}) = m_a (h_{a1} - h_{a3}) + \omega_1 m_a (h_{v1} - h_{v3}) \quad (1)$$

where m_w is the mass flow rate of cooling water and h_{w2} its enthalpy, m_a is the mass flow rate of dry air, (h_{a3} - h_{a1}) is the enthalpy change of dry air, ω_1 is the specific humidity (humidity ratio) of inlet-air in kg of water per kg of dry air, and (h_{v1} - h_{v3}) is the enthalpy change of air-borne water vapour after cooling. The humidity ratio (ω_1) can be specified directly or calculated from **[12]**:

$$\omega_1 = \frac{0.622\, p_{v1}}{P_1 - p_{v1}} \quad (2)$$

where p_{v1} is the partial pressure of water vapour and P_1 the total atmospheric pressure.

From conservation of mass, the amount of water sprayed is equal to the mass of water vapour at point 3 minus the water vapour originally in the air at point 1, i.e.:

$$m_w = (\omega_3 - \omega_1)\, m_a \quad (3)$$

where, ω_3 is the humidity ratio of air after cooling, which can also be specified directly or found from Eq. (2) if p_{v1} is replaced by p_{v3}. The partial pressures of water vapour at point 1 and point 3 can be found from the respective relative humidity (ϕ_1,ϕ_3)**[12]**:

$$p_{v1} = \phi_1\, P_{sat1} \quad (4.a)$$

$$p_{v3} = \phi_3\, P_{sat3} \quad (4.b)$$

where P_{sat1} and P_{sat3} are the saturation pressures of water vapour at the corresponding temperatures (T_1 or T_3). Neglecting pressure losses in the process, then P_3 equals P_1.

In principle, Eq. (1) can be solved to find the temperature after cooling (T_3) if the value of ϕ_3 is decided first. With conventional media-type evaporative cooling ϕ_3 can be about 90%. With fog-cooling, the rate of water sprayed can be adjusted so that air leaves the fogging system as 100% saturated However, since h_{a3}, h_{v3} and P_{sat3} all depend on the unknown value of T_3, an iterative solution procedure is required. Solving the equation by a trial-and-error method using hand-calculation would be a tedious and error-bound process. Hand calculation is also useless for performing a para-metric study of the system. Therefore, the procedure had to be programmed.

To find T_3, the program requires the user to specify the ambient temperature (T_1), the relative humidity before and after cooling (ϕ_1, ϕ_3), and the water-fog temperature (T_{w2}). For the enthalpy of liquid water (h_w) and water vapour (h_v) the following formulae are used in the program **[12]**:

$$h_w = 4.2\,T - 0.246 \quad (0.01°C<T<150°C) \quad \text{(kJ/kg)} \quad (5)$$

$$h_v = 1.8147\,T + 2501.7 \quad (0.01°C < T < 55°C) \quad \text{(kJ/kg)} \quad (6)$$

where the temperature (T) is in ^{o}C. The enthalpy change for air (h_{a3} - h_{a1}) is calculated from **[12]**:

$$h_{a3} - h_{a1} = c_p(T_3 - T_1) \quad \text{(kJ/kg)} \quad (7)$$

where c_p is the specific heat at constant pressure for air taken as 1.005 kJ/kg.K. Finally, Eq. (4) requires the saturation pressure (P_{sat}, kPa) of water vapour at a given temperature, which is obtained from the following formula (T' in Kelvin) **[12]**:

$$P_{sat} = 101.325 \exp\{70.4346943 - (7362.6981/T') + (0.006952085T' - 9.0 \ln (T')\} \quad \text{(kPa)} \quad (8)$$

2.2. The Performance Sub-model

Once the temperature after fog-cooling is determined, the improved power and heat rate of the gas-turbine can be obtained from the gas-turbine's characteristic performance curves such as those shown on Figure 2. Based on this figure, which shows the characteristic curves for the ABB GT13 gas-turbine **[13],** the turbine's power output (*PO*) and heat-rate (*HR*) at a given inlet-air temperature (*T*) can be calculated from:

$$PO = \gamma PO_s \quad \text{(kWh)} \qquad (9)$$

$$HR = HR_s/\eta \quad \text{(kJ/kWh)} \qquad (10)$$

where PO_s and HR_s are, respectively, the power output and heat rate of the gas turbine at the standard temperature of 15.6°C. For the ABB GT13, the relative power (γ) and relative thermal efficiency (η) at the desired temperature are calculated using the following formulae:

$$\gamma = 1.1007953 - 0.00684865147T - 9.6865641T^2/10^7 \qquad (11.a)$$

$$\eta = 1.0276476 - 0.0018092128T - 1.105712T^2/10^5 \qquad (11.b)$$

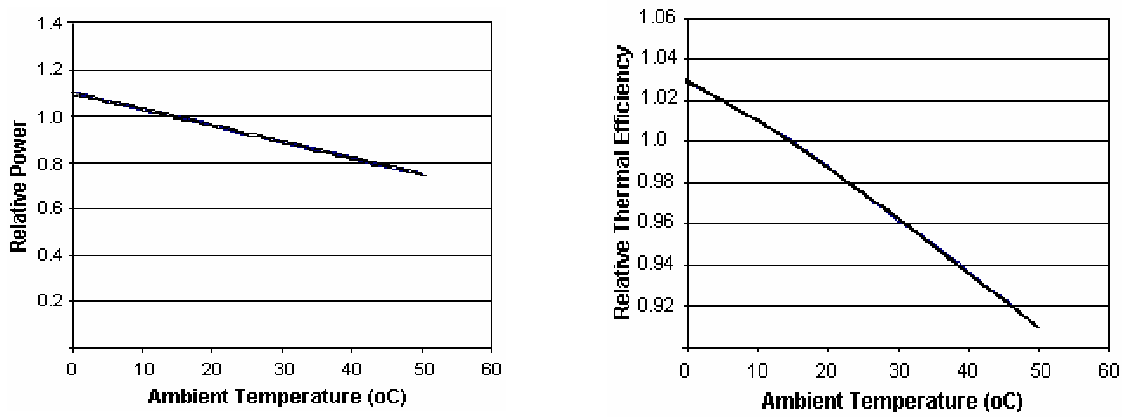
Whereas Eq. (1) forms the backbone of the inlet-air cooling sub-model, Eqs. (11.a) and (11.b) form the backbone of the performance sub-model. However, Eqs. (11), which are based on the performance characteristic curves of the ABB GT13 gas turbine, may need to be adjusted to suit other types of gas turbines. The turbine performance sub-model then obtains the revenues from fog-cooling due to the additional generation and fuel saving. The revenue from the additional generation per hour (R_e) is obtained from:

$$R_e = (PO_3 - PO_1)\, C_e \quad \text{(\$/h)} \qquad (12)$$

where PO_3 and PO_1 are, respectively, the power output with and without cooling, and C_e is the energy cost to customers ($ per kWh). The fuel saving per hour (R_f) is calculated from:

$$R_f = PO_3\,(HR_1 - HR_3)\, C_f \quad \text{(\$/h)} \qquad (13)$$

where HR_1 and HR_3 are the heat rate calculated at the ambient temperature (T_1) and the temperature after fog-cooling (T_3), respectively, and C_f is the fuel cost ($/kJ). (11.

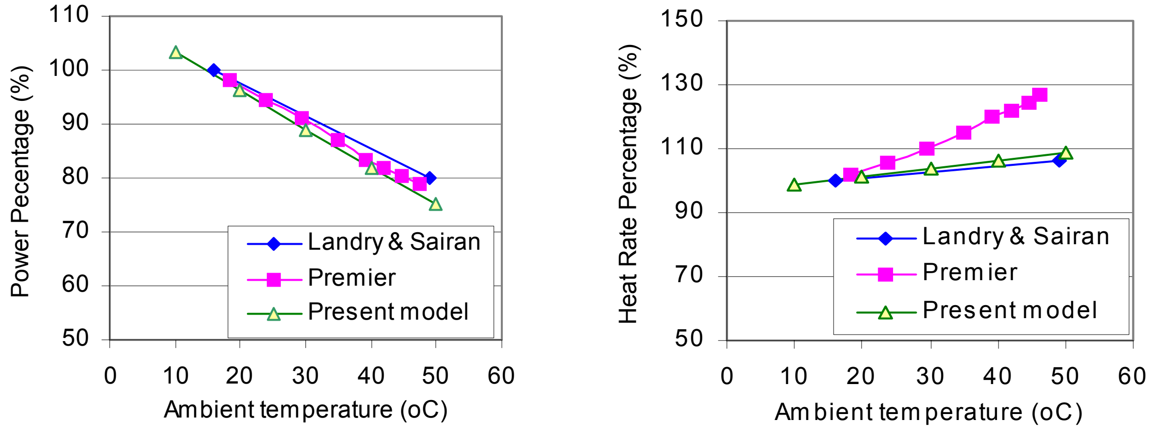


(a) (b)

FIGURE 2: Effect of ambient temperature on the power and heat-rate of the ABB GT13 gas-turbine

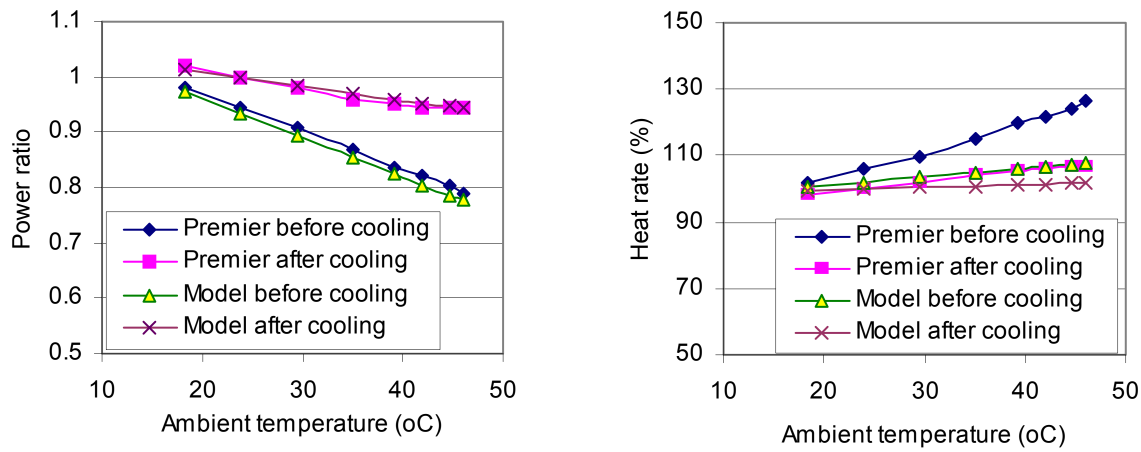
3. PROCEDURE'S VERIFICATION

Since Eqs. (11.a) and (11.b), of the gas-turbine performance sub-model, are specifically based on the performance characteristics of the ABB GT13 gas turbine, their suitability to other turbines needs to be checked. Figure 3 compares the estimation of the present procedure for the effect of ambient temperature on the gas-turbine's power and heat rate with two other estimates obtained from the literature **[7,14].** As can be seen from the figure, the procedure's estimate on the turbine power is comparable to those given by both sources. While the present estimate for the heat rate is also comparable to that given by Landry and Sairan, for an 18 MW Hitachi-GE PG-5341 turbine **[7],** it considerably deviates from that given by Premier Industries Inc **[14]** for an 83.5 MW Westinghouse turbine**.** These results indicate that Eqs. (11) can be used to give rough estimates of the effect of inlet-air cooling on the performance of gas turbines in general, but for more accuracy the equations should be tailored to the specific gas turbine being considered.



(a) (b)

FIGURE 3: Effect of ambient temperature on the gas-turbine performance: (a) power output and (b) heat-rate



(a) (b)

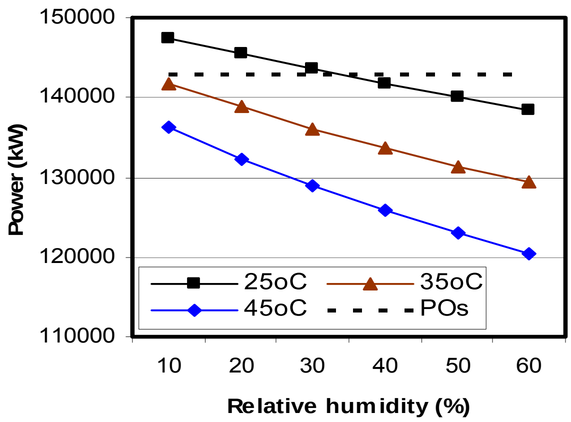
FIGURE 4: Effect of evaporative cooling on the gas-turbine performance: (a) power output and (b) heat-rate

Premier Industries **[14]** also provided data for the power and heat rate of the Westinghouse gas turbine before and after evaporative cooling. The data, shown on Figure 4, were obtained at constant inlet humidity ratio of 0.0064, but various ambient temperatures, assuming a saturation efficiency of 95%. Figure 4 compares the turbine power and heat rate with the corresponding estimates of the present procedure. Here also, our procedure correctly estimates the effect on the turbine's power, but underestimates that on the heat rate considerably.

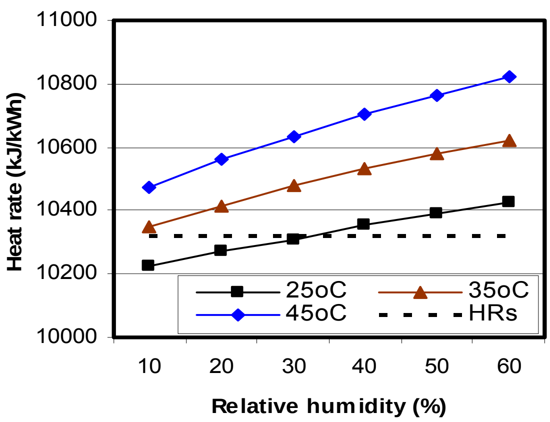
4. A PARAMETRIC STUDY

It can be seen from Eq. (1) that the main factors that limit the effect of evaporative cooling and, accordingly, the revenue from it are the ambient temperature and humidity and the temperature of the sprayed, or fogged, water. In Sudan, the air temperature and humidity vary from one region to another and, in the same region, from one season to another. To evaluate the effect of these variations on the performance of the fog-cooled gas turbine, a study has been performed using the present computer procedure. The turbine considered for the study is the ABB GT13 gas turbine for which PO_s is 143 MW, HR_s is 10320 kJ/kWh, and m_a is 500 kg/s. In the study the air and water temperatures were varied from 25°C to 45°C while the humidity was varied from 10% to 60%. Results of the computer program, for the power and heat-rate are shown on Figure 5. The figure, which also shows the turbine's power and heat-rate at the standard conditions, shows that at an ambient temperature of 35°C and 10% relative humidity, the power and heat rate of the gas turbine after fog-cooling almost match these values. However, at 45°C and 10% humidity the turbine loses about 7 MW (≈5%) even after evaporative fog-cooling.

The humidity remains low in Khartoum area for most parts of the year, which is advantageous to the cooling system. As the ambient humidity increases, the turbine's power decreases while its heat rate increases, but figure 5 shows that the effect of humidity on the heat rate is not as significant as that on the power. At an inlet-air temperature of 35°C, an increase in air humidity from 10% to 50% reduces the turbine's power by 10 MW (≈7%), but increases the heat-rate by less than 0.5%. These figures increase for the 45°C case, but decrease for the 25°C case.



(a)



(b)

FIGURE 5: Effect of ambient conditions on the effectiveness of fog-cooling: (a) power output and (b) heat-rate

Eq. (1) indicates that the performance of the fogging system can also be affected by the temperature of the cooling water-spray. To measure its effect, the temperature of the cooling-water was varied from 5°C to 35°C while the ambient temperature and relative humidity were both kept constant at 35°C and 40%, respectively. The computed results show that the air temperature after fog-cooling decreases with the water temperature, but it does so by only about 0.15°C. Although the power output increases slightly as the water temperature decreases, the difference between the power output at 35°C and at 5°C is only 156 kW (≈0.1%). The effect on the heat rate is also insignificant in the range of water temperatures considered. Regarding the water consumption, the results show that the decrease in water consumption as its temperature decreases does not exceed 0.313 m^3/h.

5. CONCLUSIONS

In Central Sudan, the temperature remains high while the humidity remains low in most parts of the year. Typical mid-day values for Khartoum area are 26°C, 18% in January, 42°C, 15% in May and 38°C, 28% in September **[15]**. The hot and dry air makes evaporative inlet-air cooling in general- and fog-cooling in particular- an effective power augmentation method for gas turbines. With the rapidly increasing demand for electricity in the country, and the highly expected shortages in power supply in the near future, retrofitting existing gas-turbines with inlet-air pre-cooling can help the power utility to reduce the gap between the demand and supply of electricity during summer and autumn times.

Evaporative fog-cooling can increase the power of the gas-turbines located in Khartoum area with an appreciable saving in fuel consumption. Although the impact of inlet-air pre-cooling on the heat-rate is usually less significant than that on the power, a greater number of the turbine annual operating hours make the heat-rate improvement also appreciable. The increased power and reduced fuel consumption are expected to generate significant revenues so that the system's pay-back period can be reduced to few months if not few weeks. Therefore, inlet-air cooling is an attractive investment opportunity for the power-generation utility.

Based on the operation data of existing or new gas-turbines, the present computer procedure can be used to determine the total annual revenues that can be made by adopting the fog-cooling system. The data required are the ambient conditions, the load-variation curve, the installation and operation costs and electricity tariff. It should be mentioned that the procedure can also be used to compare the gains of fog-cooling with those of conventional, media-type evaporative cooling by suitably adjusting the value of ϕ_3. It should also be mentioned that, in its present form, the procedure does not deal with wet compression, which requires further development of the procedure.

Finally, the present results indicate that an increase in air humidity from 10% to 60% relative humidity reduces the ABB GT13 gas-turbine's power by about 10 MW and slightly increases the heat-rate. These results indicate that air dehumidification can improve the performance of the fog-cooling system in humid environments (such as those of coastal or tropical regions) by a significant margin. However, the effect of the cooling-water temperature is insignificant and, therefore, cooling the water will not make a significant improvement to the system.

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