World Academy of Science, Engineering and Technology
International Journal of Aerospace and Mechanical Engineering
Vol:6, No:1, 2012

Comparison of Different Gas Turbine Inlet Air Cooling Methods

Ana Paula P. dos Santos, Claudia R. Andrade and Edson L. Zaparoli

***Abstract*—**Gas turbine air inlet cooling is a useful method for increasing output for regions where significant power demand and highest electricity prices occur during the warm months. Inlet air cooling increases the power output by taking advantage of the gas turbine's feature of higher mass flow rate when the compressor inlet temperature decreases. Different methods are available for reducing gas turbine inlet temperature. There are two basic systems currently available for inlet cooling. The first and most cost-effective system is evaporative cooling. Evaporative coolers make use of the evaporation of water to reduce the gas turbine's inlet air temperature. The second system employs various ways to chill the inlet air. In this method, the cooling medium flows through a heat exchanger located in the inlet duct to remove heat from the inlet air. However, the evaporative cooling is limited by wet-bulb temperature while the chilling can cool the inlet air to temperatures that are lower than the wet bulb temperature. In the present work, a thermodynamic model of a gas turbine is built to calculate heat rate, power output and thermal efficiency at different inlet air temperature conditions. Computational results are compared with ISO conditions herein called "base-case". Therefore, the two cooling methods are implemented and solved for different inlet conditions (inlet temperature and relative humidity). Evaporative cooler and absorption chiller systems results show that when the ambient temperature is extremely high with low relative humidity (requiring a large temperature reduction) the chiller is the more suitable cooling solution. The net increment in the power output as a function of the temperature decrease for each cooling method is also obtained.

***Keywords*—**Absorption chiller, evaporative cooling, gas turbine, turbine inlet cooling.

I. INTRODUCTION

GAS turbines are used for power electric generation, operating airplanes and for several industrial applications [1].The gas turbine engine consist of a compressor to raise combustion air pressure, a combustion chamber where the fuel/air mixing is burned, and a turbine that through expansion extracts energy from the combustion gases [2]. These cycles operates according to the open Brayton thermodynamic cycle and present low thermal efficiency [3] and are referred as combustion turbines.

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Usually, the rated capacities of combustion turbines are based on standard ambient air, and zero inlet and exhaust pressure drops, as specified by the International Organization for Standardization (ISO) [2]. Therefore, the air inlet conditions are: air temperature 15 °C, relative humidity 60 %, absolute pressure 101.325 kPa at sea level.

Combustion turbines are constant-volume engines and their power output is directly proportional and limited by the air mass flow rate entering the engine. Combustion turbines are constant-volume engines and their power output is directly proportional and limited by the air mass flow rate entering the engine. As the compressor has a fixed capacity for a given rotational speed and a volumetric flow rate of air, their volumetric capacity remains constant and the mass flow rate of air it enter into the gas turbine varies with ambient air temperature and relative humidity [2].

The performance of a gas turbine power plant is commonly presented in function of power output and specific fuel consumption [2], and it is sensible to the ambient conditions [4]. Thermodynamic analyses from literature show that thermal efficiency and specific output decrease with the increase of humidity and ambient temperature, but the temperature ambient is the variable that has the greatest effect on gas turbine performance [1].

The temperature ambient rise results in decrease in air density, and consequently, in the reduction of the mass flow rate. Thereby, less air passes through the turbine and the power output is reduced, at a given turbine entry temperature. Moreover, the compression work increase due the augmentation of the volume occupied by the air.

According to [4], the net power output produced by gas turbine is directly proportional to the air mass flow, it that decreases when ambient temperature increases. The work of Ibrahim, [5] shown that an increment of 1 °C in the compressor air inlet temperature decreases the gas turbine power output by 1 %.

Gas turbines have been used for power generation in several places in the world [6], [7], and each region have different climatic conditions. Furthermore, the periods of the peak electricity demand occur during the summer, when the ambient temperature is high. For example, in Arabian Gulf region the average ambient temperature presents a variation by more than 30 °C from summer to winter and this factor generate a large drop in power output during the summer [8].

Due to these severe ambient conditions, the turbine inlet air cooling is one of many available technologies to improve the performance of the gas turbine power plants by cooling the air at the compressor entry [1], [6]. Thus, the interest in the intake air cooling techniques for gas turbines has augmented in the last years, due the increasing requirement for power to a low specific investment cost [1].

Two different methods are frequently employed to obtain turbine inlet air cooling: the evaporative cooling and inlet chilling systems [7], [8]. Several works has been studied these cooling technologies as below detailed.

[1] presented a comparison between two usual inlet air cooling methods, evaporative cooler and mechanical chiller, and one new technique that uses turbo-expanders to improve performance of a gas turbine located at the Khangiran refinery in Iran. Their results showed that turbo-expander method has the better cost benefit, because it offers the greatest increase in net power and a lower payback period.

[3] performed a review of inlet air cooling methods that can be used for enhancing the power production of the Saudi Electric Company's gas turbine during summer peak hours. They concluded that the evaporative cooling system and the high-pressure fogging require a large amount of water this factor limits its use in the desert climate, the absorption chiller is an expensive system and its cost of investment isn't justifiable if it used only to improve the power output in the hour's peak. Mechanical refrigeration requires large electric power demand during the peak times, and thermal energy storage methods necessitate low electric power, but these systems need a very large storage volume. The favored alternative choose for these authors is refrigeration cooling with chilled water or ice thermal storage, the last option can produce lower inlet air temperature and requires a smaller storage volume.

[9] presented a thermodynamic assessment of some inlet air cooling system for gas turbine power plants in two different regions of Oman, and the considered techniques are evaporative cooling, fogging cooling, absorption cooling using both LiBr–H2O and aqua-ammonia, and vapour-compression cooling systems. These different cooling techniques were compared with relation their electrical energy production augmentation, as well as their impact on increasing the on-peak capacity of the considered gas turbine.

Hosseini et al. [10] modeled and evaluated an evaporative cooling system installed in gas turbines of the combined cycle power plant in Fars (Iran). Their results showed that the power output of a gas turbine, at ambient temperature of 38 °C and relative humidity of 8 %, it presents an increment by 11 MW for temperature drop of the intake air of about 19 °C.

At this context, the present work focuses on the comparison of two inlet air cooling technologies. Evaporative cooling and absorption chiller are tested at different ambient temperature and humidity conditions, and the gas turbine power output and thermal efficiency are compared.

II. Gas Turbine Unit

Fig. 1 shows the single shaft gas turbine cycle selected in this study. The compressor inlet temperature is equal to ambient temperature once the base-case neglects the cooling effect and simulates the cycle under ISO conditions (T_0 = 15 °C, P_0 = 101.3 kPa and ϕ = 60 %) and without pressure drop at inlet and exhaust ducts. Thus, the inlet pressure is given by:

$$P_0 = P_{03} \tag{1}$$

The air and combustion products are assumed to behave as ideal gases.

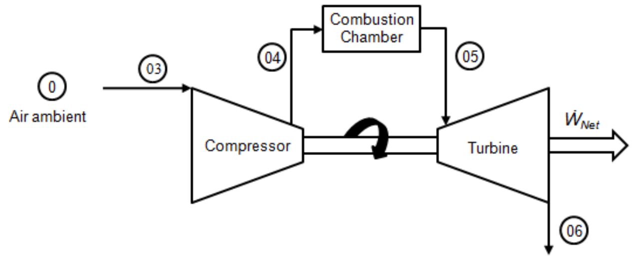

Fig. 1 Schematic of the standard gas turbine cycle

The pressure of the air leaving the compressor (P_{04}) is calculated as:

$$P_{04} = r.P_{03} \tag{2}$$

Where r is the compressor pressure ratio.

Using the polytropic relations for gas ideal and knowing the isentropic efficiency of compressor the discharge temperature (T_{04}) can be calculated as:

$$T_{04} = \frac{T_{03}}{\eta_c}\left[\left(\frac{P_{04}}{P_{03}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] + T_{03} \tag{3}$$

Where η_c is the compressor isentropic efficiency and γ is the specific heat ratio.

The compressor work can be estimated using the first law of thermodynamic as follows:

$$\dot{W}_c = \dot{m}_a.C_{pa,avg}\left(T_{04} - T_{03}\right) \tag{4}$$

Where m_a is the air mass flow rate and $C_{pa,avg}$ is the specific heat of the dry air at constant pressure, determined as a function of the average temperature across the compressor [7].

Assuming a pre-defined combustor pressure drop ($\Delta P_{Combustor}$), the combustion chamber discharge pressure (P_{05}) can be calculated as:

$$P_{05} = P_{04} - \Delta P_{Combustor} \tag{5}$$

The heat delivered by combustion chamber is determined from energy balance in it:

$$\dot{Q}_{in} = C_{pg,avg}.\left(T_{05} - T_{04}\right) \tag{6}$$

Where $C_{pg,avg}$ is the flue gas specific heat calculated as function of the average temperature across the combustion chamber [7].

By knowing the fuel gas heat value (FHV), the natural gas mass flow rate is defined as:

$$\dot{m}_f = \frac{Q_{in}/FHV}{\eta_{Combustor}} \tag{7}$$

Where $\eta_{Combustor}$ is the combustion chamber efficiency.

The turbine discharge temperature can be written as:

$$T_{06} = T_{05} - \eta_t . T_{04} \left[1 - \left(\frac{1}{(P_{05}/P_{06})} \right)^{\frac{\gamma-1}{\gamma}} \right] \quad (8)$$

Where η_T is the turbine isentropic efficiency and P_{06} is the ambient pressure.

Hence, the turbine power is equal to:

$$\dot{W}_t = \dot{m}_T . C_{pg,avg} . (T_{05} - T_{06}) \quad (9)$$

Where $\dot{m}_T$ is the total mass flow rate, it is composed for fuel and air mass flow rate:

$$\dot{m}_T = \dot{m}_a + \dot{m}_f \quad (10)$$

and $C_{pg,avg}$ is the flue gas specific heat calculated as function of the average temperature across the turbine [7].

Finally, the net power obtained from the gas turbine is given by:

$$\dot{W}_{Net} = \dot{W}_T - \dot{W}_C \quad (11)$$

The specific fuel consumption is determined as:

$$SFC = \frac{3600 . \dot{m}_f}{\dot{W}_{Net}} \quad (12)$$

An important gas turbine parameter is the heat rate (HR), calculated as:

$$HR = SFC.FHV \quad (13)$$

The thermal efficiency of the gas turbine is determined by the following equation:

$$\eta = \frac{3600}{SFC.FHV} \quad (14)$$

III. Inlet Air Cooling Technologies

Fig. 2 illustrates a simple sketch of the system herein studied, which is composed of a standard gas turbine power plant and an intake air cooler. The gas turbine power plant consists of compressor, combustion chamber and turbine. In this study, two different inlet air cooling techniques are proposed for analysis, evaporative cooling and absorption chiller.

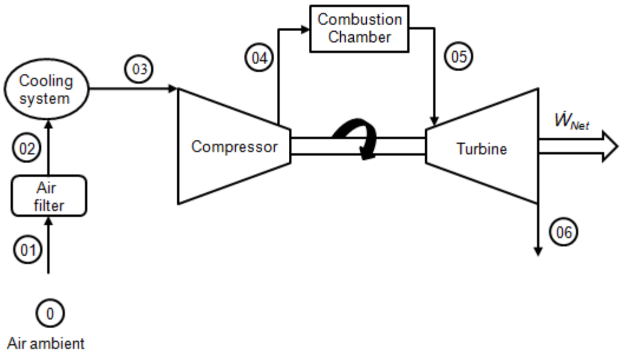

Fig. 2 Schematic of the gas turbine cycle with cooling system

The performance of the gas turbine will be evaluated with each cooling method and compared with values of the base-case. The working fluid passing through the compressor is the air, and it is assumed to be an ideal gas. While in the turbine the working fluid are the flue gases.

A. Evaporative Cooling

The evaporative cooling is most appropriated to hot dry areas, because it utilizes the latent heat of vaporization to cool ambient temperature from the dry-bulb to the wet-bulb temperature [3]. Common media types of evaporative coolers use a wetted honeycomb-like medium to maximize evaporative surface area and cooling potential, as illustrated in Fig. 3. Usually, this cooling equipment is placed after the air filter system, Fig. 2.

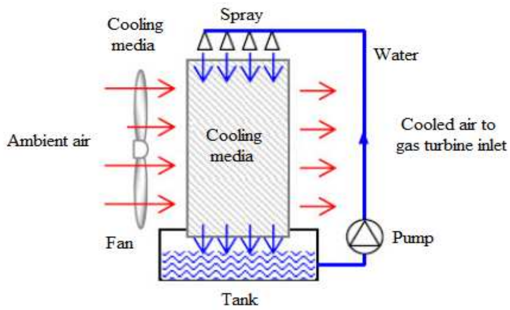

Fig. 3 Typical architecture of the evaporative cooling system

The inlet air temperature after cooling process, see Fig. 2, can be calculated as:

$$T_{03} = Tb_{02} - \varepsilon . (Tb_{02} - Tw_{02}) \quad (15)$$

Where Tb_{02} is the dry-bulb temperature, Tw_{02} is the wet-bulb temperature and ε is evaporative cooling effectiveness. The cooling load associated with the evaporative cooling system results:

$$\dot{Q}_{CL} = \dot{m}_a . C_{pa,avg} . (T_{02} - T_{03}) \quad (16)$$

Where $\dot{m}_a$ is the air mass flow rate and $C_{pa,avg}$ is the specific heat of the dry air at constant pressure, determined as a function of the average temperature across the evaporative system [7].

B. Absorption Chiller

Another option to provide gas turbine intake air cooling is the absorption chiller mechanism, as depicted in Fig. 4.

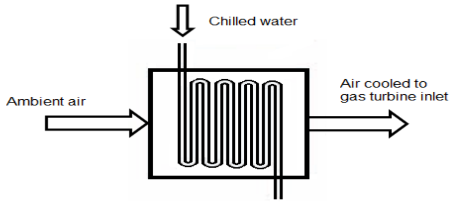

Fig. 4 Typical architecture of the absorption chiller cooling system

Absorption chiller cooling recovers heat from turbine exhaust gases and the chilled water is passed through a heat exchanger to cool the ambient air temperature. The cooling load removed from the air flowing at ambient conditions into the power plant can be calculated applied the first law of thermodynamics as follows:

$$\dot{Q}_{CL} = \dot{m}_a . \left[\left(h_{02} - h_{03} \right) - h_{f,03} . \left(\omega_{02} - \omega_{03} \right) \right] \qquad (17)$$

Where $\dot{m}a$ is mass flow rate of air, $h_{f,03}$ is the latent heat of vaporization of water, and ω_{02} and ω_{03} is the air specific humidity in the inlet and outlet of the evaporative system, respectively.

The chillers cooling different of the evaporative systems, are not limited by the ambient wet-bulb temperature [11]. The achievable temperature is restricted only by the capacity of the chilling device to produce coolant and the ability of the coils to transfer heat. Firstly, the cooling follows a line of constant specific humidity, until the saturation point is reached, and then the water of the air begins to condense, as shown in Fig. 5. The main advantage of the absorption system is that, independent of ambient air conditions, the inlet air can be cooling to a specific constant temperature and consequently increase the power output of gas turbine [3]. It is important to notice that the intake air cooling methods must be designed to avoid the formation of ice fragments on the compressor inlet or anywhere in the air intake structure. Some authors advise that the temperature drop should be greater than 5 °C [1], [7], [9].

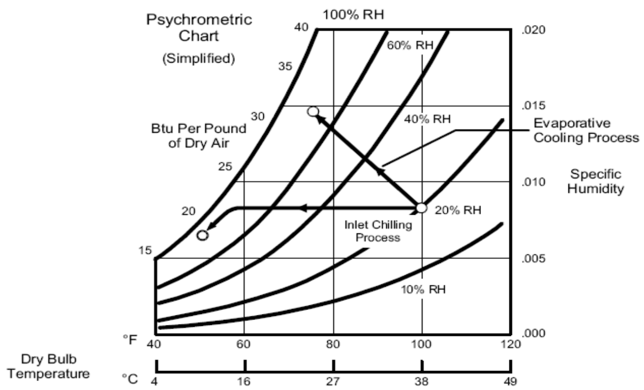

Fig. 5 Thermodynamic processes: evaporative and inlet chilling cooling systems [11]

IV. Results And Discussions

At the present work, a single shaft gas turbine is numerically simulated operating with natural gas. Table I shows the technical parameters selected for the gas turbine unit used to evaluate both the performance of the base-case (without cooling intake air) and each inlet air cooling studied method.

TABLE I
TECHNICAL SPECIFICATIONS OF THE SELECTED GAS TURBINE ENGINE

Description	Sample value
Cycle	Single shaft, simple cycle, industrial engine
Pressure ratio	11.0 [-]
Turbine inlet temperature	1,658.09 [K]
Air flow rate	141.16 [kg/s]
Isentropic efficiency of compressor	85.4 [%]
Isentropic efficiency of turbine	86.8 [%]
Combustion efficiency	99.0 [%]
Combustion chamber pressure loss	1.17 [%]
Fuel, FHV	Natural gas; 48,235.63 [kJ/kg]

Firstly, a base-case was simulated employing the ISO conditions without cooling and varying the ambient inlet temperature as shown in Fig. 6. The inlet turbine temperature was fixed at TET = 1,658.09 K.

Fig. 6 shows that as the ambient inlet temperature increases, the power output and thermal efficiency decreases in comparison with ISO rated values. On the other hand, the heat rate (see (6), (7), (12) and (13)) values elevate due to a more fuel consumption required to reach the specified inlet turbine temperature. This result shows the importance of low intake air temperature on the gas turbine performance.

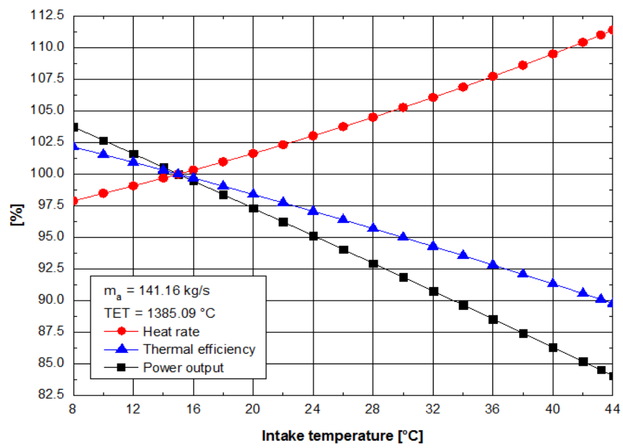

Fig. 6 Effect of inlet ambient temperature on the gas turbine performance

Fig. 7 presents the temperature decrease obtained using the evaporative cooling method as a function of the ambient intake temperature (and an ambient relative humidity fixed at 60 %). Three different evaporative cooling effectiveness values were simulated showing that a larger temperature decrease is reached when the effectiveness is higher, as expected.

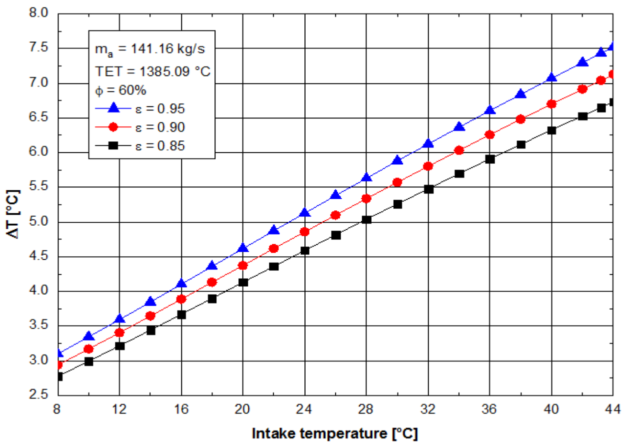

Fig. 7 Effect of evaporative cooling effectiveness on the temperature drop

A typical evaporative cooling effectiveness is ε = 0.90 that provides a temperature drop equal 6 °C when the intake temperature is 34 °C with a ambient relative humidity of 60 °C. As the wet-bulb temperature limits the application of this method, a lower ambient relative humidity condition has been tested.

The gas turbine power output is presented in Fig. 8 for both base-case and evaporative cooling inlet conditions. Note that the power output obtained is lower at ISO conditions, when the intake air is not cooled. Furthermore, the ambient dryness affects the gas turbine performance providing a higher power output level when the ambient relative humidity is lower (ϕ =18 %) in comparison with ϕ = 60 %, as shown in Fig. 8.

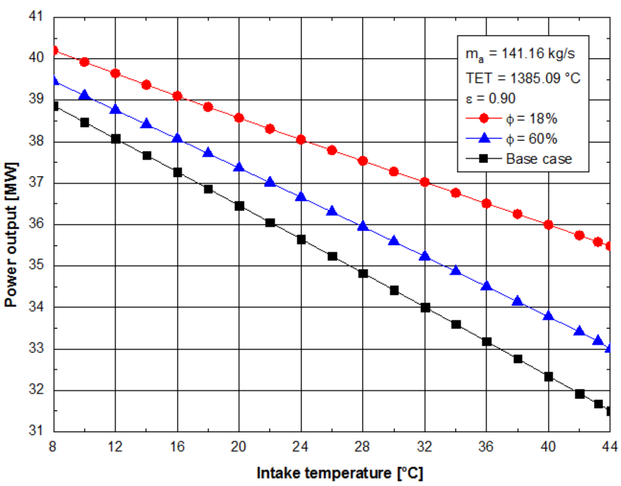

Fig. 8 Effect of ambient intake temperature on the gas turbine power output using evaporative cooling

This fact is associated with the essence of the evaporative cooling method. The ambient air passes by the cooling media following a constant enthalpy-line (Fig. 5), but the resultant temperature drop is limited by the intake air initial relative humidity.

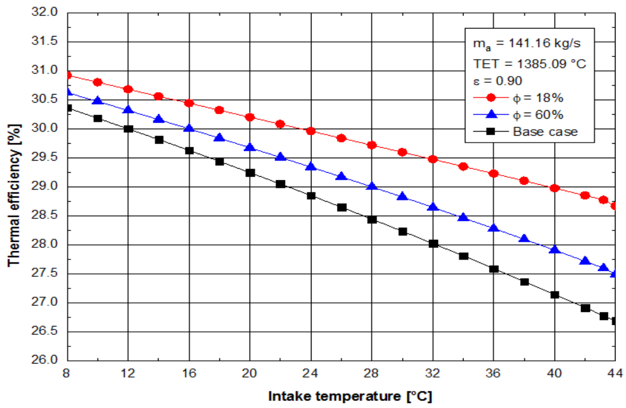

Fig. 9 Effect of ambient intake temperature on the gas turbine thermal efficiency

When the evaporative cooling technique is employed (Fig. 9), the gas turbine thermal efficiency level is higher in comparison with the base-case as occurred for the power output results. At ϕ = 60 % and air intake temperature of 34 °C, the air cooling process enhances the turbine ISO power output and thermal efficiency in 3.7 % and 2.3 %, respectively. At ϕ = 18 % the power output and thermal efficiency increase 8.4 % and 5.3 % when compared with base-case values, showing that the lower intake air relative humidity elevates the evaporative cooling performance.

Numerical simulations also included the inlet chilling method for providing compressor intake air cooling. Fig. 10 shows the temperature drop obtained employing both inlet cooling methods: evaporative and absorption chiller, at ϕ = 18 % and ϕ = 60 %.

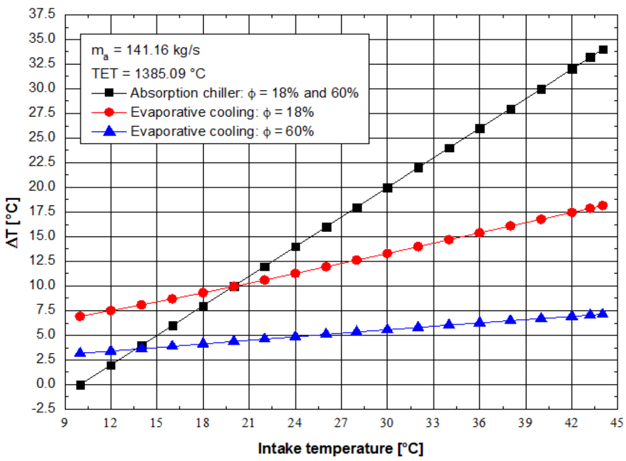

Fig. 10 Comparison between evaporative and absorption chiller inlet cooling methods

When the absorption chiller is utilized, the compressor inlet air temperature is independent of wet-bulb temperature, but there is another limitation: the compressor icing formation risk that imposes a minimum acceptable value, typically 10 °C as adopted herein.

According to temperature drop results, Fig. 10, the absorption chiller method reaches a better cooling effect in comparison with evaporative cooling mainly when the ambient intake temperature is higher (up to 20 °C). At ambient temperature lower than 20 °C, the evaporative cooler temperature drop is larger, due to the fixed inlet compressor temperature specified in the absorption chiller (T_{03} = 10 °C), see Fig. 2.

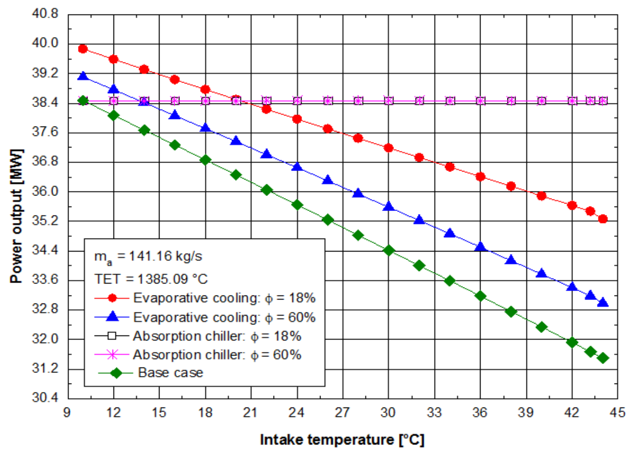

Fig. 11 Effect of ambient intake temperature on the gas turbine power output using evaporative and absorption chiller cooling

Fig. 11 presents the gas turbine power output results obtained by evaporative cooling and inlet chilling techniques. The evaporative system has a better performance in comparison with base-case although its results depends on the ambient relative humidity (at ϕ = 18 % and T_0 = 30 °C, the power output is about 1.5 MW superior to ϕ = 60 % and T_0 = 30 °C).The power output gain is even more considerable when the absorption chiller is applied. For example, at T_0 = 30 °C, the increment is about 4.2 MW when compared with base-case value. This same behavior is presented by the gas turbine thermal efficiency results, as shown in Fig. 12.

However, the evaporative cooling method provides a higher thermal efficiency when the ambient intake temperature is lower than 20 °C at relative humidity equal to 18 %.This fact occurs because at these conditions, the evaporative system attains a larger temperature drop in comparison with absorption chiller technique, where the minimum compressor inlet temperature was pre-established.

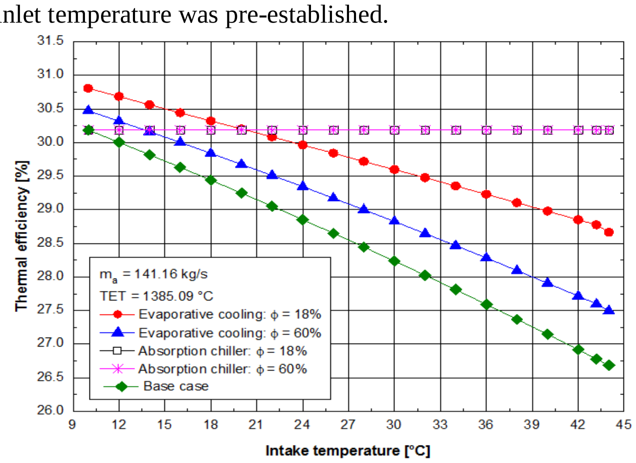

Fig. 12 Effect of ambient intake temperature on the gas turbine thermal efficiency using evaporative and absorption chiller cooling

It is also verified that the advantage of evaporative system at relative humidity equal to 60 % is only noticed for low inlet ambient temperature, inferior to 14 °C, when the cooling requirements are not pertinent.

V. Conclusion

A numerical simulation of a single shaft gas turbine utilizing two different inlet cooling techniques is presented. While the base-case (at T_0 = 34 °C and ϕ = 18 %) provided a gas turbine power output equal to 33.59 MW, the evaporative cooling brought an increment of 8.4 % and the absorption chiller represented a power output gain of 12.7 %.

Results showed that both methods improve the power output and thermal efficiency when compared with base-case (gas turbine operating under ISO conditions). Nevertheless, the evaporative cooling method was limited by the ambient wet-bulb temperature, representing a suitable solution at low ambient relative humidity inlet conditions, Fig. 12. On the other hand, the absorption chiller reached a larger temperature drop at different ambient conditions. Therefore, if the exhaust gas energy is available, this method represents a better option once it can be utilized independent of the ambient relative humidity level.

Acknowledgments

The authors are grateful to VSE (Vale Soluções em Energia) by the financial support for the development of this work.

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