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CONSTRUCTAL APPROACH ON THE FEASIBILITY OF COMPRESSED AIR TEMPERATURE CONTROL BY EVAPORATIVE COOLING IN GAS TURBINE POWER PLANTS

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Abstract. Summer higher atmospheric air temperatures affect the performance of gas turbine power plants. To prevent in such conditions diminishing of the produced mechanical power there are already available two technologies based on water injection. The first, known as “fog cooling”, is a simple technology to reduce the intake air temperature by direct evaporative cooling based on spraying liquid water at the axial compressor entrance. The “inter-stage water spraying” technology is the second one, where the liquid water is gradually injected along the axial compressor. This paper aims to evaluate the potential of these two technologies for improving the overall performance of the gas turbine power plants. To investigate the limits and conditions in which the real axial compressor's design may evolve, the numerical results are calculated based on the Constructal Theory by modeling the compressor as an ensemble of adjacent elemental isothermal compressors that ”...provides easier access to the imposed currents that flow through it”. Compressed moist air temperature control within each of the elemental compressor is done based on the evaporative cooling principle. Numerical results, for the first technology, agree well with test bench values obtained for the TV2-117A version of a gas turbine converted by COMOTI from kerosene to gaseous fuel.

***Key words*:** Evaporative cooling, Isothermal compression, Gas turbines, Constructal Law.

1. INTRODUCTION

Performances and the long lifetime of the modern gas turbine power plants (GTPPs) make them well suited for cogeneration groups. The International Standards Organization (ISO) considers the following characteristics of the atmospheric air for the gas turbines design: $T^* = 15^0$C, $P^* = 101.3$ kPa and $\phi^* = 60\%$. During the summer time the atmospheric air temperature increases while the air density decreases, resulting in the GTPP's produced power reduction. Depending on the gas turbine type employed, the atmospheric air temperature may change the operating point of both the compressor and the turbine, such that 10^0C increases in the ambient temperature reduces the power of the GTTP with 5–13% and the efficiency with 1.5–4% respectively [1]. Together, these particularities of the gas turbines operation, the requirements for competitiveness in the field of electricity and the obligation to comply with the EU Directive 2010/75 provisions, result in a great challenge for finding technical solutions for compensating for power losses during the summer without building additional power plants.

To avoid shortcomings of the GTPP’s performance during the summer time, there are already available some technologies for cooling down the intake air such as: (1) the evaporative cooling systems that mixture atmospheric air with liquid water that evaporates producing lower temperature moist air, and (2) indirect cooling systems continuously cooling upstream of the compressor the intake air by thermal interaction within a heat exchanger [2].

The evaporative cooling of the intake air is done in several ways: (i) traditional system forcing the flow of the atmospheric air through a soaked porous media; (ii) evaporative cooling spraying liquid water directly into the atmospheric air stream entering the axial compressor (“inlet fogging”), with overspray variant (or wet compression); (iii) inter-stage evaporative cooling, when the liquid water is gradually injected along the axial compressor, allowing thus for the optimization of the compressor functioning since in this way it can be injected the desired quantity of liquid water at the exact position for better controlling the compressed gas temperature.

In the indirect cooling systems case, the intake air cooling occurs upstream of the compressor, within heat exchangers that may use low temperature water or some refrigerant fluid as cooling fluid. Detailed analysis of the intake air cooling technologies was presented by Melino [2], while various other studies were developed for specific locations with high summer temperature in Brazil by Celis *et al*. [3], in Saudi Arabia by Ibrahim and Vamham [4], and in Oman by Dawoud *et al*. [5].

Accounting the complexity of the two-phase flow process and the polydisperse spraying, Bhargava *et al*. [6] consider that there is a poor understanding of the phenomenology of wet compression in the compressor stages of the gas turbine. Thus, the authors state that the efficiency of wet compression is influenced by a variety of factors such as the droplets diameter, gas pressure and temperature, and the mass flow rate of injected liquid water. When inter-stages water spraying is used, the injection position along the compressor becomes also an important parameter. Anurov [7] studied the dynamics of the evaporation process of water droplets along the 13 stages compressor of the GT-009 gas turbine installation, by considering various parameters such as the injected water flow rate, water injection position towards the airstream and the diameters of water droplets.

To contribute with the decision-making on directions for developing the existing air cooling technologies in GTPPs, and for better understanding the fundamentals and its maximum potential for increasing the efficiency, this paper presents a constructal approach on the evaporative cooling process employed to control the temperature of the compressed air in GTPPs.

2. FUNDAMENTALS OF EVAPORATIVE COOLING IN THE GTPPs

Evaporative cooling is one of the most simple and cheap intake air cooling technologies applied on gas turbine industry, compared with nowadays sophisticated techniques (mechanical chiller or absorption ones, thermal storage etc.). Temperature rise attenuation of intake air mass flow is achieved due to contact with water which vaporizes. Theoretically, the minimum temperature that can be achieved is the wet bulb temperature. Practically, this temperature is difficult to be achieved and the temperature drop is dependent on both the equipment and the environmental conditions.

2.1. “Fog cooling” technology

Intake air cooling technology by spraying water mist form was applied to a gas turbine since 1980 [8]. Normally, fine water droplets (with a diameter of less than 30 µm) sprayed into the intake air stream (Fig.1) evaporates until entering the compressor. Depending on the ambient conditions, the water mass flow rate injected towards the airstream is about 1–2% of the air flow rate.

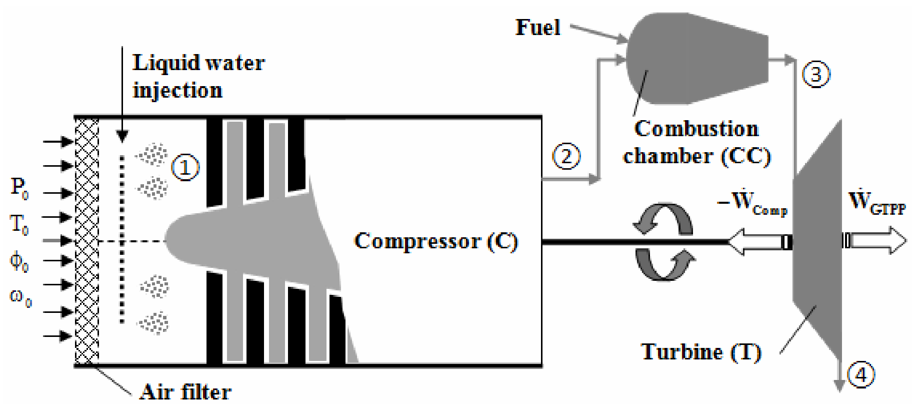

Fig. 1 – Physical configuration of a GTPP employing the "fog cooling" technology with complete vaporization before the admission into the compressor of the liquid water sprayed into the intake.

Advantages of this type of method for gas turbine performance improvement are: low installation costs; reduced pressure drop on compressor intake; high efficiency; inexpensive technological reconfiguration; can be located either upstream or downstream of the air filter.

2.2. "Inter-stage water spraying" technology

Technology that involve water injection between the axial compressor stages (Fig. 2) presents special nozzles to ensure a homogeneous jet with droplets having a diameter below 10 μm and a very short compressor residence time, in the range of 15...30 ms. This technology can provide a higher power increase than the water injection technique into the compressor intake. Also, this method could remove compressor working away from stall limit, ensuring a more stable operation of the entire gas turbine engine.

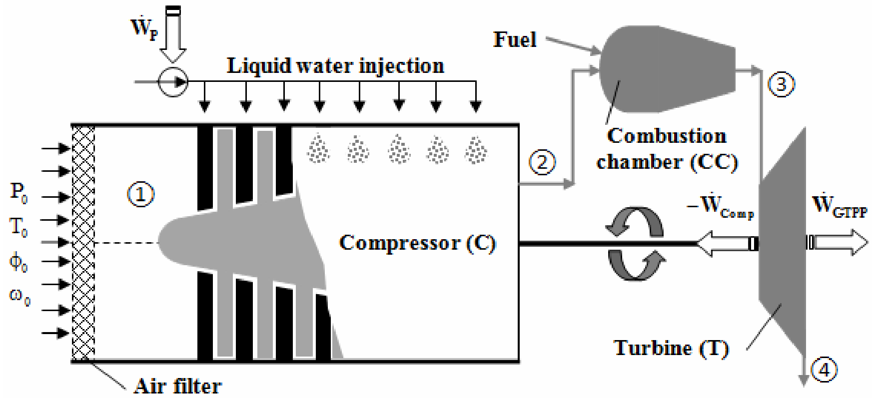

Fig. 2 – Physical configuration of a GTPP employing the "inter-stage water spraying" technology.

It should be outlined the idea that the compressor characteristic will be changed for the wet compression evolution, compared to dry compression; moreover the wet compression depends on the compressor overall geometry and the size of the water droplets.

3. CONSTRUCTAL APPROACH OF COMPRESSED AIR TEMPERATURE CONTROL

To provide "greater and greater access to the currents that flow through" the internal volume of a gas turbine power plant's (GTPP's) axial compressor, the existing configurations shown in Figs. 1 and 2 are replaced by a "globally easier flowing configuration" as shown in Fig. 3, where the liquid water injection is considered to develop along the compressor. Then, in order to quantitatively address the problem, the internal configuration of the air compressors in Figs. 1 and 2 is replaced by a repetitive pattern of many adjacent elemental compression stages.

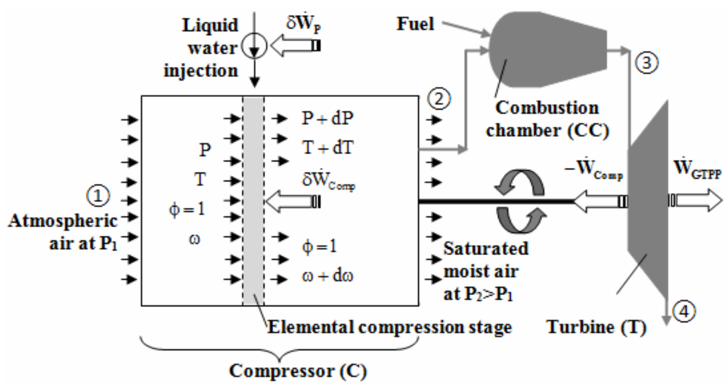

Fig. 3 – Adjacent elemental compression stages within the internal volume of the GTPP's axial compressor.

4. MATHEMATICAL MODEL

The mathematical model is based on the equations of mass and energy conservation and on the second law of thermodynamics applied to the control volumes representing the elemental compression stages in Fig. 3. The combustion chamber (CC) and the turbine (T) are modeled based on their global characteristics, as usual for gas turbine Brayton thermodynamic cycle. The model employed in this study determines the characteristic parameters by focusing firstly on the reversible functioning and then on irreversible operation. The dimensionless ODE system to approach the reversible operation is written down as follows in the equations (1), (2), (3) and (4):

$$\frac{d\tilde{T}'}{d\tilde{P}} = \left[\frac{0.622\tilde{P}}{\left(\tilde{P}-\tilde{P}_{sat}\right)^2} \Delta\tilde{s}_{fg} + \frac{k_a - 1}{k_a}\frac{1}{\tilde{P}} \right] \Bigg/ \left[\frac{1}{T} + \omega \frac{ds_w'^{vap}}{d\tilde{T}'} + \frac{0.622\tilde{P}}{\left(\tilde{P}-\tilde{P}_{sat}\right)^2} \Delta\tilde{s}_{fg} \frac{d\tilde{P}_{sat}}{d\tilde{T}'} \right], \tag{1}$$

$$\frac{d\omega'}{d\tilde{P}} = \frac{0.622\varepsilon\phi\tilde{P}}{\left(\tilde{P}-\tilde{P}_{sat}\right)^2} \left(\frac{d\tilde{P}_{sat}}{d\tilde{T}'}\frac{d\tilde{T}'}{d\tilde{P}} - 1 \right), \tag{2}$$

$$\frac{\delta\left(\tilde{W}'_{Comp}/\tilde{m}_a\right)}{d\tilde{P}} = -\left[\left(1 + \omega' \frac{d\tilde{h}_w^{vap}}{d\tilde{T}'} \right) \frac{d\tilde{T}'}{d\tilde{P}} + \Delta\tilde{h}_{fg} \frac{d\omega'}{d\tilde{P}} \right], \tag{3}$$

$$\frac{\delta\left(\tilde{W}'_p/\tilde{m}_a\right)}{d\tilde{P}} = -\left(\frac{P^*}{\rho_w^{liq} c_{pa} T^*} \right) \left(\tilde{P} - \tilde{P}_0\right) \frac{d\omega'}{d\tilde{P}}, \tag{4}$$

while for the irreversible case the dimensionless ODE system is given by equations (5), (6), (7) and (8):

$$\frac{d\tilde{T}''}{d\tilde{P}} = \left[\frac{0.622\tilde{P}}{\left(\tilde{P}-\tilde{P}_{sat}\right)^2} \Delta\tilde{h}_{fg} - \frac{1}{\eta_{s,C}} \frac{\delta\left(\tilde{W}'_{Comp}/\tilde{m}_a\right)}{d\tilde{P}} \right] \Bigg/ \left[1 + \omega'' \frac{d\tilde{h}_w^{vap}}{d\tilde{T}''} + \frac{0.622\tilde{P}}{\left(\tilde{P}-\tilde{P}_{sat}\right)^2} \Delta\tilde{h}_{fg} \frac{d\tilde{P}_{sat}}{d\tilde{T}''} \right], \tag{5}$$

$$\frac{d\omega''}{d\tilde{P}} = \frac{0.622\varepsilon\phi''\tilde{P}}{\left(\tilde{P}-\tilde{P}_{sat}\right)^2} \left(\frac{d\tilde{P}_{sat}}{d\tilde{T}''}\frac{d\tilde{T}''}{d\tilde{P}} - 1 \right), \tag{6}$$

$$\frac{\delta\left(\tilde{S}''_{gen,Comp}/\tilde{m}_a\right)}{d\tilde{P}} = \left[\left(\frac{1}{\tilde{T}''} + \omega'' \frac{d\tilde{s}_w^{vap}}{d\tilde{T}''} \right) \frac{d\tilde{T}''}{d\tilde{P}} + \Delta\tilde{s}_{fg} \frac{d\omega''}{d\tilde{P}} - \frac{k_a - 1}{k_a}\frac{1}{\tilde{P}} \right], \tag{7}$$

$$\frac{\delta\left(\tilde{W}''_p/\tilde{m}_a\right)}{d\tilde{P}} = -\left(\frac{P^*}{\rho_w^{liq} c_{pa} T^*} \right) \left(\tilde{P} - \tilde{P}_0\right) \frac{d\omega''}{d\tilde{P}}, \tag{8}$$

where k_a, ρ_w^{liq}, c_{pa} and $\eta_{s,C} = \tilde{W}'_{Comp}/\tilde{W}''_{Comp}$ are the ratio of specific heats of dry air, the liquid water density, the dry air's specific heat at constant pressure and the compressor isentropic efficiency, respectively. $\Delta\tilde{h}_{fg}$ and $\Delta\tilde{s}_{fg}$ represent the dimensionless specific enthalpy and entropy of vaporization for water, while $\tilde{h}_w^{vap}$ and $\tilde{s}_w^{vap}$ the dimensionless specific enthalpy and entropy of the water vapor.

Dimensionless temperature, pressure, mass flow rate, specific enthalpy and entropy and mechanical power are given by: $\tilde{T} = T/T^*$, $\tilde{P} = P/P^*$, $\dot{m} = \dot{m}/\dot{m}_a^*$, $\tilde{h} = h/\left(c_{pa}T^*\right)$, $\tilde{s} = s/c_{pa}$, and $\tilde{W} = \dot{W}/\left(\dot{m}_a^* c_{pa} T^*\right)$. (*) superscript refers to the ISO characteristics of the atmospheric air for the gas turbines design, while the prime (´) and the double prime (´´) superscripts indicate whether the numerical values of the parameters have been determined for the reversible or irreversible functioning. The subscripts (*a*), (*Comp*), (*sat*) and (*w*) refer to the dry air, to the axial compressor, saturation conditions and water respectively.

5. RESULTS AND DISCUSSION

Based on technical characteristics of the TV2-117A [9] version of a gas turbine converted by COMOTI from kerosene to gaseous fuel, the numerical calculations were developed based on the ODEs systems already presented that have been numerically solved for $1 \le \tilde{P} \le \Pi = 7$ based on the 4th order Runge-Kutta procedure. More, turbine inlet temperature was considered the same for all the cases. Successive refinements of the dimensionless pressure step were used to verify the convergence of the numerical results.

Table 1

Numerical values of the GTPP's estimated performances

Temperature control type	Compressor (C)			CC	Turbine	GTPP
	$\tilde{T}_2''$	ω_2''	$\tilde{W}_{Comp}''$	λ	$\tilde{W}_T''$	$\tilde{W}_{GTPP}''$
	GTPP's ISO estimated performances					
No control of the compressed air temperature	1.877	0.0063	-0.877	4.105	1.545	0.668
GTPP's summer time estimated performances						
No control of the compressed air temperature	2.007	0.0208	-0.860	4.253	1.449	0.589
″Fog cooling″ technology employed	1.047	0.0249	-0.855	4.134	1.483	0.628
Indirect evaporative cooling at the compressor aspiration and "inter-stage water spraying" according to the Constructal approach	1.047	0.0871	-0.724	2.812	1.658	0.934

The numerical results obtained for the evaluation of the GTPP performance when using the types of compressed moist air temperature control mentioned in Table 1 are shown graphically in Fig. 4. The curves (1) and (2) represent in Fig. 4, respectively, the temperature variation of the compressed moist air when a temperature control is not used either under ISO conditions or for the atmospheric air characteristics $T_1 = 35^{\circ}\text{C},\ P_1 = 101.3\,\text{kPa}$ and $\phi = 60\%$ of the summer time.

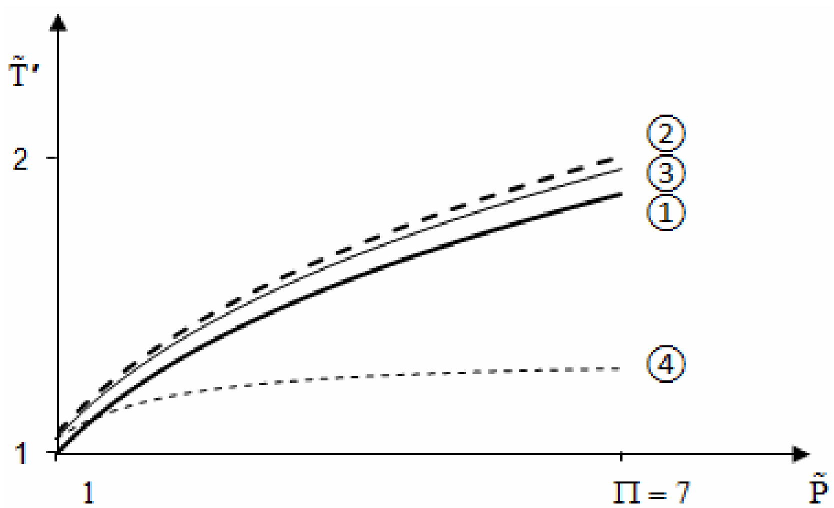

Fig. 4 – Temperature variation of the compressed moist air when using different types of temperature control.

When the “fog cooling” technology is considered for improving the summer time GTPP performance, the compressed moist air temperature variation is represented in Fig. 4 by the curve (3). Graphically represented by the curve (4) in Fig. 4 are also the numerical results obtained when approaching based on the Constructal Law the GTPP’s summer time performance with indirect evaporative cooling at the compressor aspiration and “inter-stage water spraying”.

6. CONCLUSIONS

The numerical results obtained in this work show, at least from a fundamental point of view, the effectiveness of compressed air temperature control by evaporative cooling in gas turbine plants.

There is a continuous compressed moist air temperature increase tendency when a temperature control is not used either under ISO conditions or for the summer time air characteristics, nor when the "fog cooling" technology is employed to improve the summer time GTPP performance. Meanwhile, the results obtained based on the Constructal approach, with indirect evaporative cooling at the compressor aspiration and “inter-stage water spraying”, show comparatively that during most of the compression process a temperature of the compressed moist air 45% lower.

Providing a “globally easier flowing configuration”, the Constructal approach with indirect evaporative cooling at the compressor aspiration and “inter-stage water spraying”, shed some light on the potential of growth in the mechanical power delivered by the GTPP of 45.81% and a simultaneous reduction of 2.26% in the fuel specific consumption.

Numerical results on the entropy generation due to the compressor functioning confirm the realism of the adopted assumptions and the adequacy of the mathematical formulation of the proposed model for numerical simulation purposes.

Application of water injection in each compressor stages involves miniature holes drilled on stators and a sophisticated water injection system. The experiments regarding water injection through TV2-117A intake, reconfigured from kerosene to methane showed not only a power augmentation but also NO_x emissions reduction similar to the work presented in [10]. Thus the introduction of water into the gas turbine intake at idle generates a reduction of averaged turbine inlet temperature of about 5 ^{0}C and also NO_x emissions fall by approximately 4 ppm. Idle regime becomes important for “stand-by GTTP” that is used only when energy peaks occur since they can work as little as one hour but the pollutant emissions should be low all the time.

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