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The Influence of Inlet Air Cooling and Afterburning on Gas Turbine Cogeneration Groups Performance

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Additional information is available at the end of the chapter

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1. Introduction

Usually, cogeneration is defined as combined production of power and thermal energy from the same fuel source, represented by natural gas, liquid fuel, refinery gas, etc. In conventional energy production the efficiency is approximately 40 %, but through cogeneration it can reach even 90 %. Fuel supply and increased performance requirements, environment concerns, continuously variable market conditions have contributed to the development of the gas turbines. The performances, exploitation costs, safety in operating conditions have made these installations to be selected for cogeneration processes.

2. State of art

Gas turbine systems operate on the ideal thermodynamic cycle (consisting in two isentropic and two isobars) represented by Brayton cycle. The real Brayton cycle consists in quasiadiabatic expansion and compression processes, but unisentropic, and the heat transfer processes are not isobar processes, due to flow pressure losses. In addition, the air and hot gases are not perfect gases and not have the same flow rates. Brayton cycle thermal efficiency depends on: compression ratio; ambient temperature; air temperature at turbine inlet; compressor efficiency and turbine components efficiency; blade cooling requirements; increased performance systems (exhaust gases heat recovery, intercooling, intake air cooling, afterburning imple-

mentation, fluids injection – water/steam, etc.). The main parameters that define the operating thermodynamic cycle of gas turbine installations (usually disclosed by the suppliers in catalogues) are the temperature at the gas turbine inlet (T_3) and the compression ratio. Generally, gas turbine manufacturers declare performances without taking into consideration the inlet and outlet pressure losses. Gas turbine installations performances are affected by the variation of these parameters as follows [2]: temperature increase at the gas turbine inlet leads to an increase in power and efficiency; the efficiency becomes maximum at a given value of the compression ratio (in $T_3 = \text{const.}$ hypothesis); there is a value of the compression ratio for which the power is maximum (T_3 and compressor intake air flow rate remains constant). The productivity of a gas turbine cogeneration group depends on the quantity of heat recovered from the turbine exhaust gases (approximately 60-70 % from the fuel energy). This is achieved by adding a heat recovery steam generator in order to supply hot water or steam. Determining factors in total efficiency of the cogeneration group are the gas turbine outlet temperature and the temperature at the stack of the heat recovery steam generator. The combination temperature at the gas turbine inlet – compression ratio determines the outlet temperature. The gas turbine, being located at the upstream of the heat recovery steam generator, significantly influences the cogeneration group performances. The air is induced by the gas turbine compressor in ambient conditions imposed by the location of the cogeneration plant. Compressor inlet temperature and intake air density dictates mechanical work required by the compression process, the fuel and quantity of fuel to be used in order to obtain the necessary temperature at the gas turbine inlet (T_3). Consequently, output power, efficiency, exhaust gases mass flow and outlet temperature (respectively the quantity of heat recovered) are influenced by ambient conditions [3]. The location of the gas turbine cogeneration plant imposes climatic conditions and requires adequate technical solutions in order to ensure performances. Generally, for cogenerative applications, the gas turbine is designed to operate in standard conditions, established by the International Standards Organization and defined as ISO conditions: 15 °C, 1.013 bar and 60 % humidity. During summer season air temperature rises and its density decreases, leading to a decrease in the intake air mass flow; consequently decreases and power output because it is proportional to the intake air mass flow rate. Without taking supplementary measures, both gas turbine output power and efficiency drop. In the scientific literature there are various papers that deal with the gas turbine's performance dependence of the intake air temperature variation [3-10]. In [4] it is shown that: an increase of 10 °C at the compressor inlet reduces the gas turbine outlet power with 18%; in comparison with the operation during winter season, the increase of ambient temperature leads to a decrease in gas turbine plants power output with 25-35%, also leading to an average increase of the consumption of 6%. The effect of intake air temperature over the performances differs from one gas turbine to another, but, generally, aeroderivative gas turbines are more sensitive to this phenomenon than the industrial gas turbines [5]. During summer season, when the days are long and hot, the power requirements increase for the residential spaces ventilation, offices, store rooms, etc. Additional energy consumption can be ensured by starting other backup groups, or compensating the loss of power through various other methods. The usual compensation methods of power loss are [6, 7]: compressor inlet air cooling (pre-cooling), intermediate cooling (intercooling), using recovery cycle. Mainly there are two basic com-

pressor inlet air cooling methods: evaporative cooling (with evaporative media cooling or water injection in the inlet air-fogging); refrigeration system cooling [8]. For a 79 MW gas turbine, equipped with a fogging cooling system, the researches conducted at Mashhad (in Iran) showed that during a day, the maximum increase in power is achieved in the afternoon, when the temperature is higher and relative humidity is lower [9]. Inlet air cooling systems analysis in order to be applied to a gas turbine V94.2, in terms of efficiency increase, led to the conclusion that the fogging cooling system meets the design requirements and leads to an increase in power output of approximately 6 MW [10]. With the help of GT PRO software the performances of a 100 MW gas turbine model were analyzed, for a various types of inlet air cooling systems, and it had been reached that a decrease of air temperature of 1 °C (in the 25-35 °C interval) leads to an increase in power output of approximately 0.7 MW [11]. For a gas turbine cycle, with intermediate cooling (intercooler), the decrease of inlet air temperature causes the output power to rise and the intercooling leads to a 5-9% gain of power and a 8% reduction in fuel consumption [12]. Reduction of fuel consumption represents a priority both for industrial gas turbine manufacturers and also for the civil aviation. The search is on for new materials that meet the requirements imposed by the higher strains of the gas turbines [13] and also the development of new technologies, including technological transfer from aviation domain to power generation domain. Thus, in aviation, afterburning is used in order to increase traction of supersonic engines. The introduction of afterburning into cogenerative applications leads to an increase in flexibility and global efficiency of the cogeneration group. Afterburning application is possible due to the fact that exhaust gases at turbine outlet have a 11-16% (volumes) content of oxygen [14]. The afterburning installation, located between the gas turbine and the heat recovery steam generator, interacts with the gas turbine but influences the heat recovery steam generator operation especially [14, 15]. To increase the performance of gas turbine cogeneration groups research focused specifically on [16]: increase in burning temperature; increase in compression ratio; improving the methods of design, cooling and burning technologies, and also advanced materials; technological transfer from the aviation domain in the industrial gas turbine domain and conversion of aviation gas turbine (with outdated lifetime) to energy conversion; integrated systems (combined cycles, compressor inlet air cooling, intercooling, turbine exhaust gases heat recovery, afterburning implementation, chemical recovery, etc.). Following the direction displayed in the field, the chapter integrates data from scientific literature with research developed at INCDT COMOTI Bucharest, regarding gas turbine inlet air cooling and afterburning application, as base methods for increasing performances and flexibility of cogenerative group.

3. Influence factors and methods of increasing performances in gas turbine cogenerative groups

For a combined cycle (considering as variables ambient temperature, gas turbine outlet temperature and stack temperature) it is shown that the dominant factor in global efficiency rise is stack temperature [17]. Obtaining a high efficiency involves the optimization of the entire cogenerative plant (gas turbine, afterburning installation, heat recovery steam generator, etc.).

The efficiency must be maintained even at partial loads (even under 50%) in variable conditions modification. In general, although the target is obtaining a maximum efficiency, nevertheless an adequate flexibility to process requirements is desired, the afterburning installation contributing to this.

3.1. Influence factors

Ambient parameters (humidity, pressure, temperature) can vary significantly depending on geographic location and season, affecting air density and implicitly the gas turbine cogenerative group performances. In the past, the effect of air humidity was neglected but the increase in gas turbine cogenerative groups power and the introduction of water/steam in the combustion chamber made this effect to be reconsidered. Thus, some authors [18] consider that air relative humidity (even at temperatures higher than 10 °C) has a neglectable influence over the gas turbine output power (as the other performance parameters). This leads to the fact that in some calculus (especially when the results are presented in correlation to ISO conditions) the variations in atmospheric humidity and pressure to be neglected. Others consider that due to the fact that water content modifies thermodynamic properties of inlet air (density, specific heat), at certain gas turbines (depending on specific processes) the performances may increase when humidity rises and in the case of some gas turbines the performances may decrease in the same conditions [19]. However, the increase in relative humidity leads to a significant reduction of NO_x emissions [20].

Ambient pressure is defined by the conditions from plant location, altitude modification leading to air density modification and implicitly to power output variation. Thus, 3-4% losses occur for each 304.8 m (1000 ft) rise in altitude [21].

Power and efficiency of the gas turbine group decrease along with ambient temperature, in figure 1 linear approximate variations being presented. Specific fuel consumption increases with the ambient temperature rise [22].

Gas turbines operate on a wide variety of gaseous fuels (natural gas, liquefied natural gas-LNG, liquid petroleum gas-LPG, refinery gas, etc.) and liquid fuels (kerosene, no. 2 diesel, jet A, etc.). Using a certain type of fuel for the gas turbine has a profound impact both on the design and also on material selection. Usage of liquid fuels imposes: ensuring burning without incandescent particles and residues on the combustor and turbine; reducing hot gas corrosive effect due to aggressive compounds (sulphur, lead, sodium, vanadium, etc.); resolving pumping and pulverization (filtering, heating, etc.) issues. In case of using gaseous fuels, a simpler solution is presented due to their higher thermal stability, higher heating power, lack of ash and smut. However, in order to ensure pressure level (required by the gas turbine, afterburning installation, etc.), water and various impurities elimination implies a control-measuring station for the gaseous fuels used (natural gas in the case of cogenerative plant 2xST18 – Figure 2). Although the main fuel for the operation of gas turbine cogenerative groups is natural gas, the economic rise and environmental requirements issued an alternative. The gas turbine can be designed to operate on a variety of fuels, but the rapid transition to other fuel operation, without machine damage or exceeding the level of emissions, still remains an issue subjected to study.

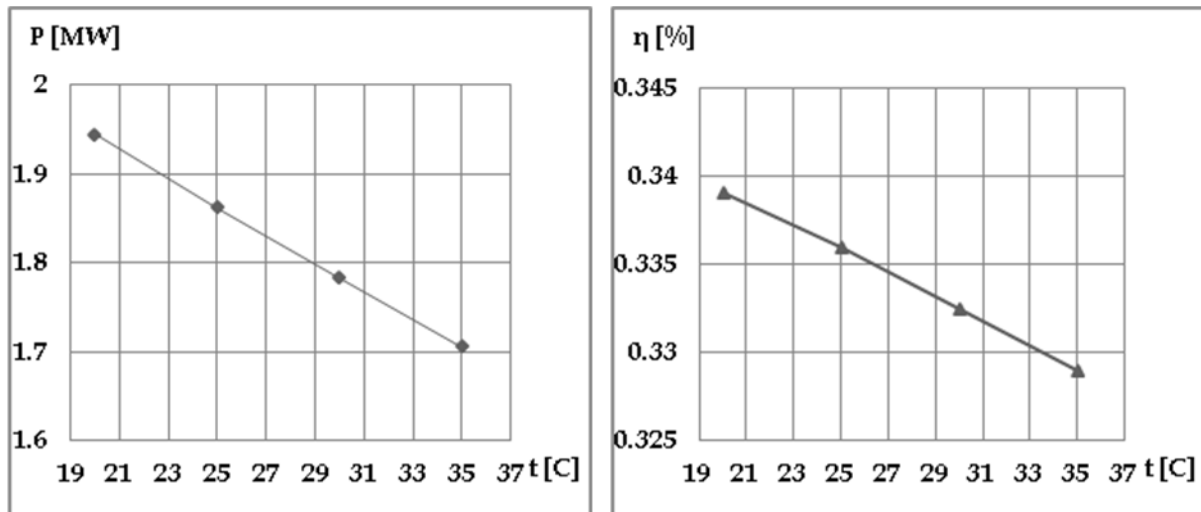


Figure 1. Power output (left) and efficiency (right) variations versus ambient temperature, at a simple gas turbine cycle

Interchangeability at gaseous fuel gas turbine cogenerative groups, represents the ability to change a fuel with another one, without affecting the application or the equipment in which the gaseous fuel is burned [15]. At a constant fuel composition (in the case that the holes through which the fuel passes to the burner have fix dimensions), the quantity of heat delivered by the burner is proportional with the mass flow and heating power. When the composition varies and it is the problem of replacing a fuel with another equivalent, Wobbe index (after John Wobbe-engineer and mathematician) is used as comparison criteria. It is defined as ratio of the lower heating value (LHV) and the square root of relative fuel density respectively, in relation to air (d_{rel}):

$$Wo = LHV / (d_{rel})^{0.5} \quad (1)$$

$$d_{rel} = \rho_{comb} / \rho_{air} \quad (2)$$

Thus, two gaseous fuels (with different chemical compositions), with the same Wobbe index, are interchangeable and the quantity of heat delivered to the equipment is equivalent at the same fuel supply pressure. In order to take into account the fuel temperature, a Wobbe temperature corrected index can be used. According to [23], two gaseous fuels are interchangeable if the following relation is satisfied:

$$\frac{\Delta p_2}{\Delta p_1} = \left(\frac{Wo_2}{Wo_1} \right)^2 \left(\frac{A_1}{A_2} \right)^2 \quad (3)$$

where Δp_1 and Δp_2 represent the overpressures of gas 1 and gas 2 respectively, W_{O_1} and W_{O_2} – Wobbe indices of fuels 1 and 2, A_1 and A_2 – gaseous fuel injection nozzle surface areas.



Figure 2. Cogenerative plant 2xST 18 – Suplacu de Barcau (left) and afterburning installation (right) [14, 15]

Thus, the validation criteria of adequate replacement of one fuel with another equivalent fuel are given by: self-ignition, flame temperature (with a high influence on NO_x emissions forming), flame speed, flashback, efficiency, NO_x and CO emissions, flue gas dew point, etc. Resolving gas turbine cogenerative group fuels interchangeability, by developing high performance alternative fuel burning technologies, especially hydrogen, will have a major impact over system and environment efficiency. Thus, the studies conducted on more fuels (H_2 , CH_4 , C_3H_6 , C_6H_6 , CH_3OH) revealed that [24]: hydrogen and methyl alcohol have the same higher maximum efficiency than other fuels in the same operating conditions; hydrogen fuel has the lowest specific fuel consumption in comparison with other fuels, followed by methane, propen, benzene, and finally methyl alcohol; in the reheating cycle the increase in thermal efficiency is lower than the increased in the intercooling cycle; hydrogen fuel is ideal promising fuel in the gaseous plant which has greater thermal efficiency and greater improvement in the performance of modified gas turbine power plant occurred with intercooling and heat exchanger rather than simple and reheat cycle.

From the point of view of reusing aviation gas turbines for industrial purposes, the possibilities of using liquid fuels are limited, leading to the development of new technologies on gaseous fuels. Thus, the transition of a TV2-117A gas turbine engine from liquid to a gaseous fuel, in order to benefit from landfill gas energy value, has been conducted in many stages: transition from liquid fuel (kerosene) operation to gaseous fuel (natural gas) operation, thus obtaining TA2 gas turbine engine; the transition of TA2 gas turbine engine from natural gas operation to landfill gas operation, thus obtaining TA2 bio gas turbine engine. In order to obtain the two gas turbine engines (TA2 and TA2 bio), numerical simulations were conducted in CFD environment, constructive modifications and gas turbine test bench experiments in order to

validate adopted solutions [15]. In this way, TA2 gas turbine engine was integrated in the structure of afterburning installation test bench facilities, from INCDT COMOTI Bucharest (figure 3).

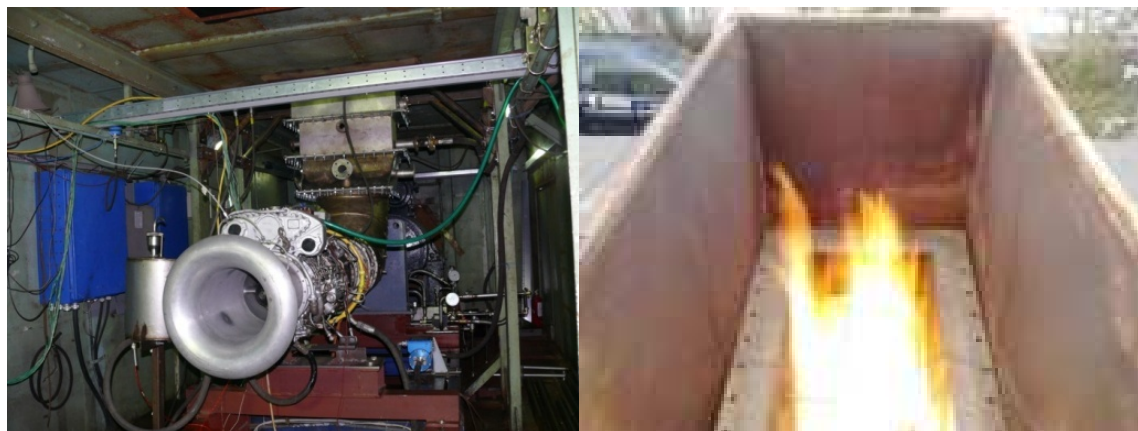


Figure 3. TA2 gas turbine engine mounted on the stand (left) and afterburning installation (right)

Efficiency and reliability are two major parameters that are taken into account since the beginning of a new gas turbine cogenerative group design. In order to obtain a higher efficiency (in balance with cost and reliability), the design team must reach a balance between burning temperature rise and compression ratio, special material selection and complicated cooling systems, customer's specification, etc. Based on a continuum dialogue with Siemens customers, as far back as the predesigned stage, a 36 MWe SGT-750 gas turbine was elaborated [25]. It can be used both in cogenerative applications as well as driving different equipment, stable operation at partial load but also allows the transition to another fuel (operates on dual fuel). Since 2002, Siemens began focusing attention to reliability, so that they give up the high pressure tambour from the recovery boiler (usually used in order to prevent high thermal tensions, a long period of time is imposed in order to reach a certain temperature). Regarding efficiency, flexibility and emission reduction at gas turbine cogenerative groups, important steps were achieved towards: integrating low NO_x burners, lifecycle was analyzed in order to increase efficiency, maintenance interval was enlarged and the transition from one fuel to another was improved at polifuel groups.

3.2. Methods of increasing performances

The development of gas turbine application in various regions of the globe, encouraged researchers to find new methods of increasing performances and to apply new cooling technologies to compressor inlet air, specific to the location. Actually the advantages of inlet air technology application are represented by power losses prevention, losses that occur when ambient temperature exceeds $15\text{ }^\circ\text{C}$ (ISO design temperature), and fuel usage efficiency. In general, the studies of selecting a new method of cooling air take into account each method (see chapter 1) and compare them with a reference case [8]. Evaporative cooling systems imply lower investments, operating and maintenance costs than refrigeration systems but the

increase in gas turbine performances is lower. Evaporative cooling system is based on the transition of an air flow through a water soaked environment, system efficiency depending on the surface area of the water soaked environment exposed to air and exposure time. The system is efficient in low humidity regions. In the case of fogging system, the demineralized water is sprayed in the gas turbine inlet air flow, through high pressure nozzles (70 – 200 bar). Figure 4 shows a fogging system, located downstream of the filtering system, in a gas turbine. The air induced through the filtering system reaches the suction chamber, where the water, sprayed in fine droplets (<30 microns), is evaporated in the air flow and produces a cooling effect. In general, the water injected mass flow required represents 2 % from the inlet air, depending on the ambient parameters (temperature and humidity).

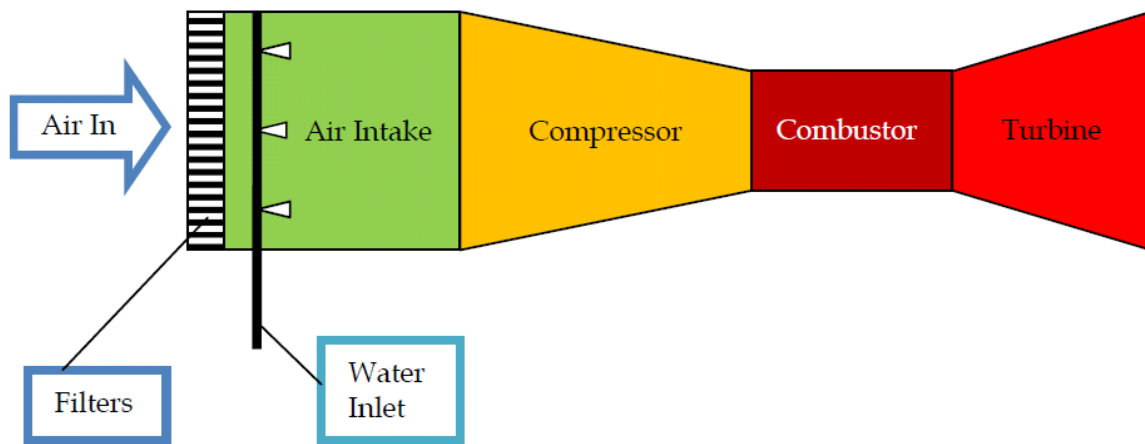


Figure 4. Fogging system scheme, located downstream of the filtering system

Water injection system (with fog) has the advantage of lower pressure drops, higher efficiency, lower refurbishment costs but requires demineralized water, parasite loads occur due to water injection pumps and the control system is complex. Fogging systems, increasingly popular in time, are capable of cooling gas turbine intake air to the temperature of the wet-bulb thermometer, being more efficient than the ones with evaporative media. Although significant investments are required, refrigeration cooling systems (mechanical, absorption, storage) can practically maintain any temperature of gas turbine intake air, recommended in variable humidity regions. In general, fogging system are designed to operate in such a way that the formed droplets should evaporate until entering the compressor, in order to avoid blade damage. In some cases overspray phenomenon can occur (a certain amount of water would not evaporate until entering the gas turbine). In this way overspray phenomenon is similar with water injection between compressor stages (intercooling phenomenon occurring). In certain operating regimes it is possible that evaporative cooling (upstream of the compressor) combined with overspray phenomenon (intercooling) lead to compressor aerodynamic instability (although intercooling effect is added to evaporative cooling) [26]. Combining overspray phenomenon with the usage of a low power heating fuel, steam injection in the combustion chamber or a high level of blade wear can lead to catastrophic results for the gas turbine. Some authors [27] consider that at gas turbine cogenerative cycles, intercooling is

recommended, in order to reduce high pressure compressor stage power consumption. Thus, the factors that need to be taken into account when adopting a new technology of cooling the inlet air are [28]: plant configuration (gas turbine engine can be used within an open cycle, a cogenerative cycle, a combined cycle with high operation – intermittent or basic); the amount of compressed air per kW (requiring a small amount of compressed air the cooled air required decreases and, implicitly, exploitation costs decrease), the ambient air enthalpy at given conditions by the design (selecting a high enthalpy of air per kg of dry air leads to a large cooling system, with high exploitation costs); air cooling temperature (for each gas turbine engine model exists an optimum cooling temperatures at given environment conditions).

Although the integration of an afterburning installation (figure 2 and 3) can lead to an increase in cost (with approximately 10-15 % from the cost of the heat recovery steam generator) certain advantages are highlighted: an increase in steam amount at the heat recovery steam generator; thermal energy can be managed more easily; steam amount efficiency, depending on technological process requirements; the heat recovery steam generator can operate even at gas turbine shutdown; it can compensate ambient parameter variations; in some cases can contribute to NO_x emissions reduction by interacting with gas turbine emissions; can burn other fuels usually inadequate to gas turbines (biogas, furnace gas, flammable gases resulted from gasification, etc.).

4. Theoretical research developed at INCDT COMOTI Bucharest

At INCDT COMOTI Bucharest, theoretical research approached issues concerning: processes regarding water spray in gas turbine intake; intercooling; influence of air cooling over gas turbine performances; thermogasodynamic processes from the afterburning installation's burner.

Thus, in order to understand the phenomenon of cooling air at water spraying through an impact body nozzle [29], numerical simulations in CFD environment were conducted, with the working domain consisting in air (gas) and water (liquid). In the first version the impact body (a cone with 1.2 mm base and generator lines) was not taken into account. In the second version the cone was positioned, firstly, at 0.4 mm towards the pipe's end, then moved at 0.8 mm, and 1 mm respectively. Sprayed water reaches in the calculus domain (a cylinder of 1m in diameter and 2.5 m in length) through a 1.2 mm diameter and 12 mm length pipe positioned along the cylinder's symmetry axis. In order to capture as accurately as possible the turbulence phenomenon that occurs in the cone and pipe area, at the same time with water flow sprayed around the cone, the mesh was refined (figure 5).

Also, along the pipe and cone walls boundary layers were created, in order for an accurate capturing of the flow near the walls. Water is sprayed into the atmosphere with an axial speed of 15 m/s, as droplets of 100 μm diameter, having a temperature of 288 K. Boundary conditions for the cone and pipe are of smooth adiabatic wall type. The turbulence model used was k-ε. Work methodology was based on the model described in [30]. Numerical simulation results

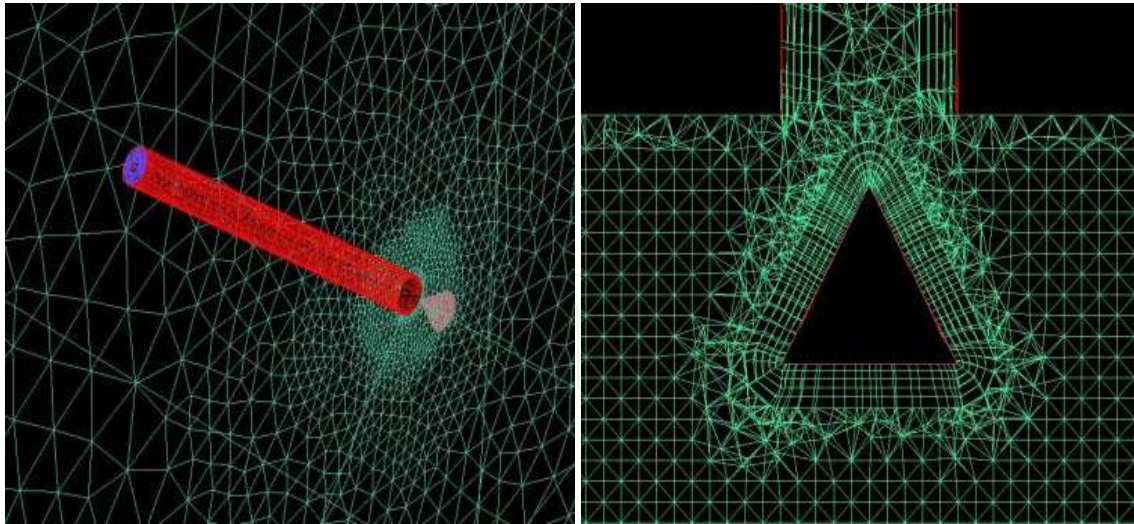


Figure 5. Calculus mesh at the inlet zone of sprayed water in the domain (left) and around the cone (right)

are presented in figures 6 and 7, highlighting the spray without impact body and with impact body positioned at 1 mm distance from the pipe outlet.

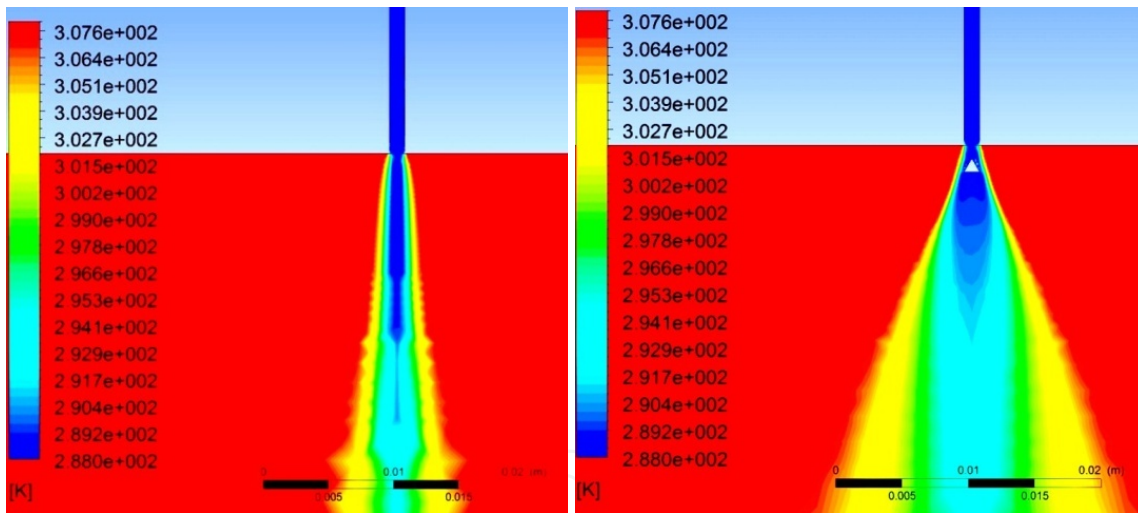


Figure 6. Detail regarding temperature field at water output: without cone (left) and with the cone positioned at 1 mm distance (right)

Figure 6 reveals that cone insertion leads to pronounced water jet flaring. Cone position modification has a minimum effect over the water spray flare angle. For a better observation of sprayed water jet along the calculus domain, 300 mm in diameter planes were created (comparable with TA2 gas turbine engine intake-figure 3) along the symmetry axis corresponding to z axis coordinate of 0.5 m, 1 m, 1.5 m, 2 m, and 2.5 m respectively.

Figure 7 shows that in case of numerical simulations without the cone, the water jet is directed along the symmetry axis, while in case of impact body numerical simulations the water

jet is more dispersed, as expected. In the case of numerical simulations without the cone, the average temperatures decrease more rapidly.

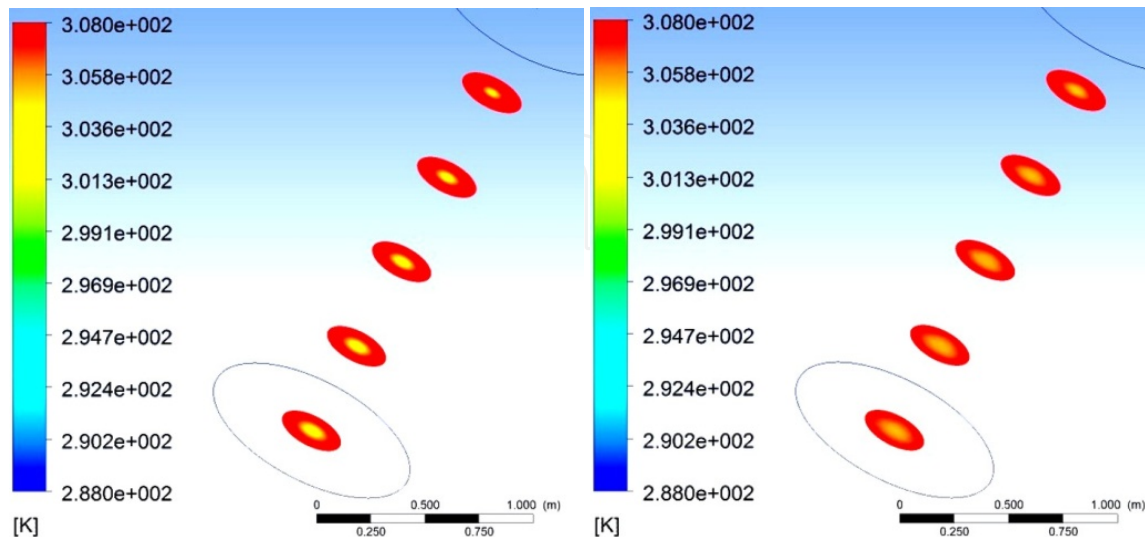


Figure 7. Temperature field in different cross planes: without cone (left) and with the cone positioned at 1 mm distance (right)

Compressor performances are influenced by flow steadiness at the rotor outlet, more specifically if the flow angle varies from hub to shroud than the flow within the diffuser will be unsteady and will lead to separations on the blade vane. Also, the flow at the tip of the blade and at the centrifugal rotor outlet is intensely distorted and unsteady. In order to quantify the impact of intercooling on a two stage compressor, a CFD study was conducted in which the second stage inlet temperature was reduced. Performance impact was monitored, in comparison with the case in which between the two compression stages no cooling was applied. Thus, two CFD analyses were conducted, in which the total inlet second rotor temperature was considered 460 K, for the case without cooling between the compression stages and 313 K, for the case with cooling.. The calculus mesh for this case is a structured mesh and has approximately 1 million elements. The walls are considered to be adiabatic, waterproof and velocity at wall is considered to be zero. In order to improve calculus time, a single channel has been considered (rotor consisting in 15 blades and 15 splitters), using the periodicity function. For spatial discretization was used a second order scheme. A compressible flow was considered in calculus, the governing equations being written as Reynolds Averaged, time and mass averaged. Shear Stress Transport $k-\omega$ was considered as turbulence model. SST $k-\omega$ model is based on tangential tensions transport. With the help of this turbulence model, accurate results and separation zone dimension (that forms under the influence of high pressure gradients) can be obtained. The results are presented in figures 8-10.

Analyzing the CFD simulations results, it can be observed that along with temperature reduction at rotor inlet, streamlines in the rotor indicate that a recirculation zone is forming on the splitter. This shows that flow angles in the rotor have changed, fact that led to boundary

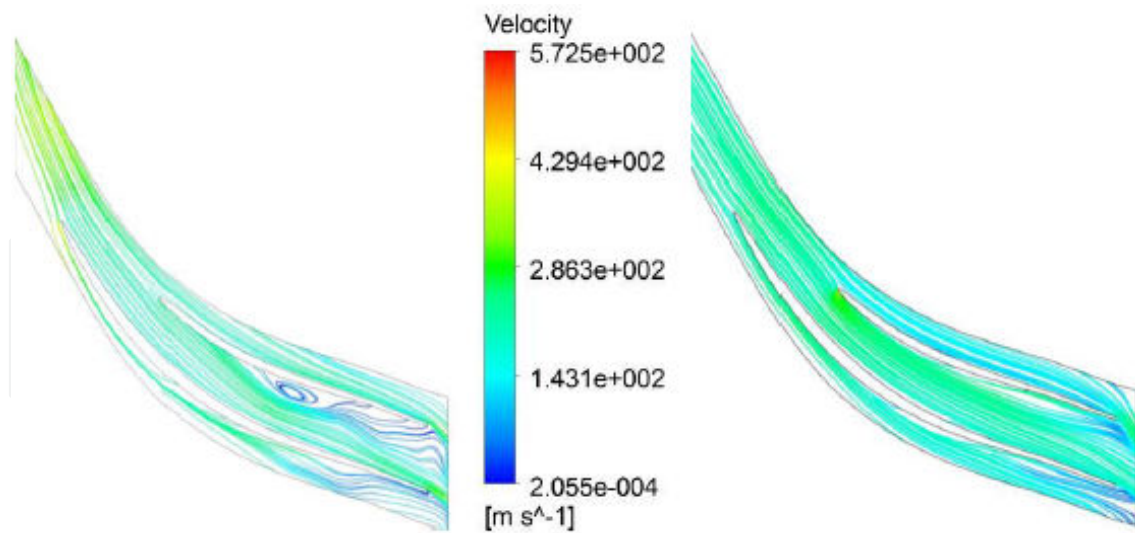


Figure 8. Streamlines, at 50 % of blade height, measured from the hub, in the case of cooling (left) and in the case without cooling (right)

layer separation in the case of intercooling (figure 8). Also, it can be observed (in meridional plane) the fact that Mach number rises at the rotor outlet, from 0.88 in the case without intercooling to 1.055 in the case with intercooling (figure 9).

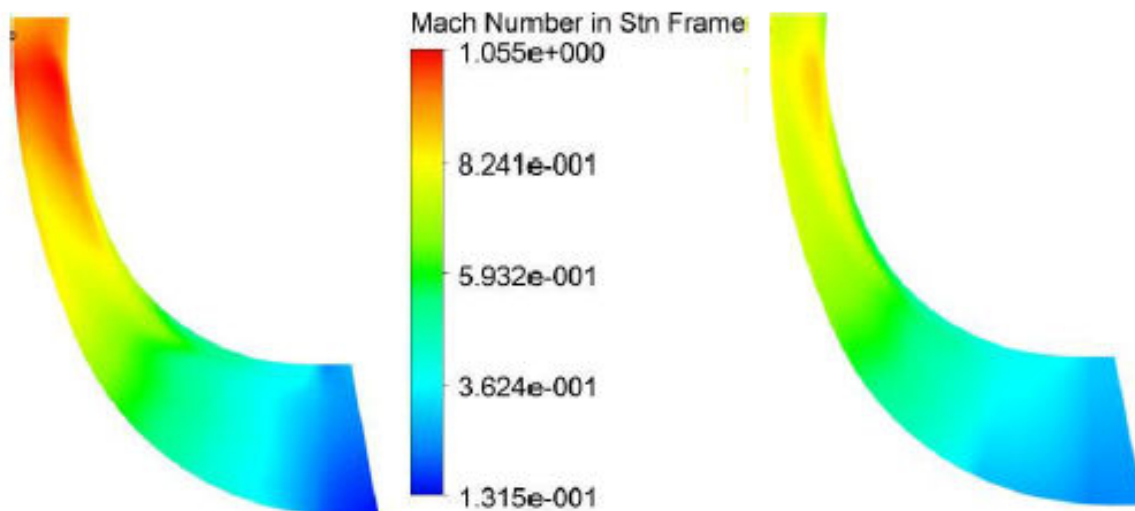


Figure 9. Mach number in meridional plane, in the cooling case (left) and in the case without cooling (right)

Another difference can be observed also in outlet rotor total pressure, the pressure increases in the case of intercooling (figure 10). Thus, at centrifugal rotor outlet, in the case of intercooling, an increase of 0.2 in Mach number can be observed, reaching 0.938. This indicates a transonic regime at rotor outlet. Corroborated with the fact that the flow angle at rotor outlet reached 12.72° from 17.713° , it means that the existing stator must be redesign.

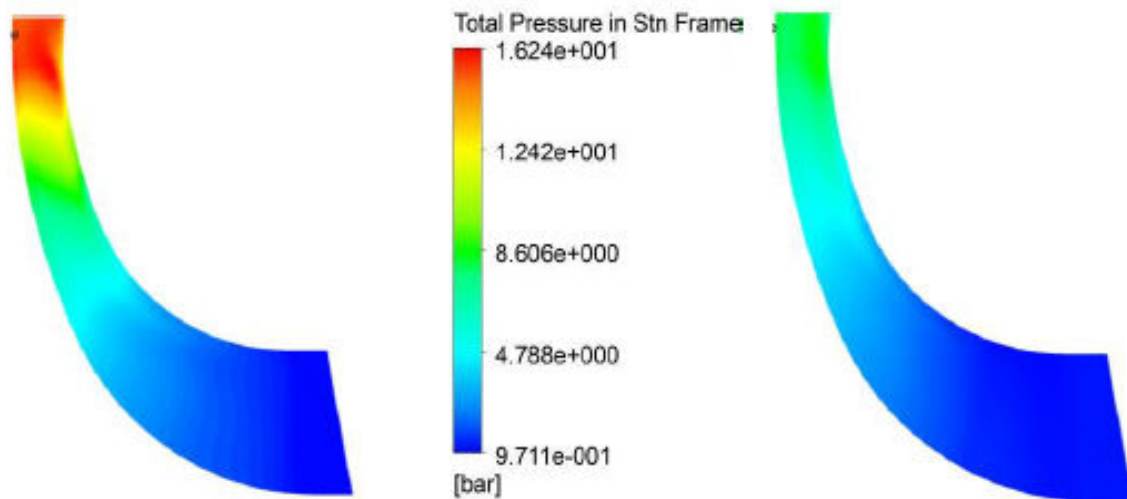
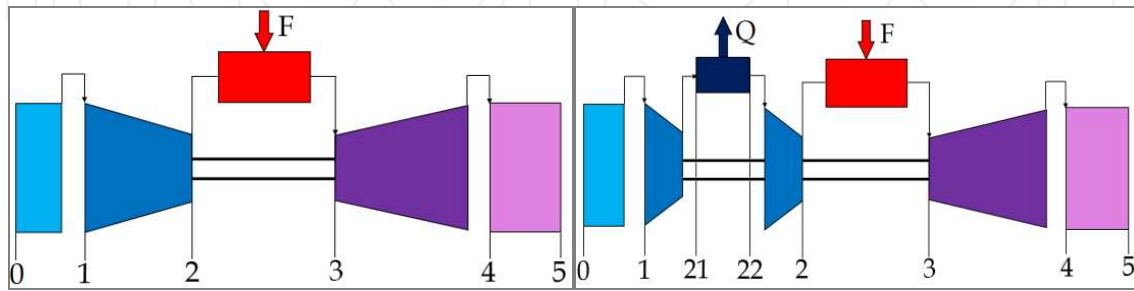


Figure 10. Total pressure in meridional plane, in the cooling case (left) and in the case without cooling (right)

In general, the performances of a turboshaft engine is compared with the ones given by ideal cycle calculus, thermodynamic analysis including thus, the ideal gas turbine work range. Gas temperature at turbine inlet, as gas turbine outlet section and outlet pressure, are equal in the case of ideal and real gas turbine [31].

Thus, the main configuration to be taken into account for thermodynamic analysis is the one for a monorotor turboshaft engine, with two high and low pressure compressor stages, coupled with a turbine that includes both gas generator stages, as well as power shaft supply stages. For performance calculus, the above configuration is considered (figure 11-left) and intercooling configuration (between compressor low and high pressure stages – figure 11 right). The heat exchanger, that provides intercooling, decreases air temperature in the high pressure compressor inlet section at 40 °C (313 K). Intercooling application decreases the mechanical work consumed by the compressor, without affecting the mechanical work produced by the turbine, leading to an increase in the mechanical load available at the output shaft [32]. In thermodynamic analysis, for the two gas turbine engine configurations presented above, a series of parameters were imposed for operational purposes. The calculus method used is in accordance with [33]. Common conditions to both configurations refer to: ambient temperature, whose variation influences gas turbine performances (shaft power, thermal efficiency, specific fuel consumption); gas temperature in the turbine inlet section, having the same value in all the cases presented, does not vary with ambient temperature, parameter imposed by turbine alloy properties; turbine outlet pressure, in order to ensure exhaust, in the context of pressure losses on the exhaust; burned gases pressure at the exhaust outlet so that the gases can pass through the heat recovery steam generator; coefficients of pressure, speed, energy losses with compressor and turbine efficiencies, and compression ratio (in initial configuration), low pressure compressor (in intercooling configuration) respectively are considered constant. In addition, for the configuration that includes intercooling, the following parameters are imposed: temperature at high compressor inlet, the same for all calculus cases, that allows

the use of a low temperature resistant material and allows to obtain a high compression ratio at the same mechanical work consumed on the second compressor; the pressure loss coefficient in the heat exchanger; air pressure at combustion chamber inlet, that derives from total compression ratio imposition. In these conditions, the gas turbine output power variation depending on the inlet air temperature (for the various compression ratio), for simple configuration and intercooling and for different compression ratios, is presented in figure 12.



(0 – intake inlet; 1 – compressor inlet (low pressure compressor); 21 – heat exchanger inlet; 22 – high pressure compressor inlet; 2 – combustion chamber inlet; 3 – turbine inlet; 4 – exhaust inlet; 5 – exhaust outlet)

Figure 11. Gas turbine engine configurations – general schemes

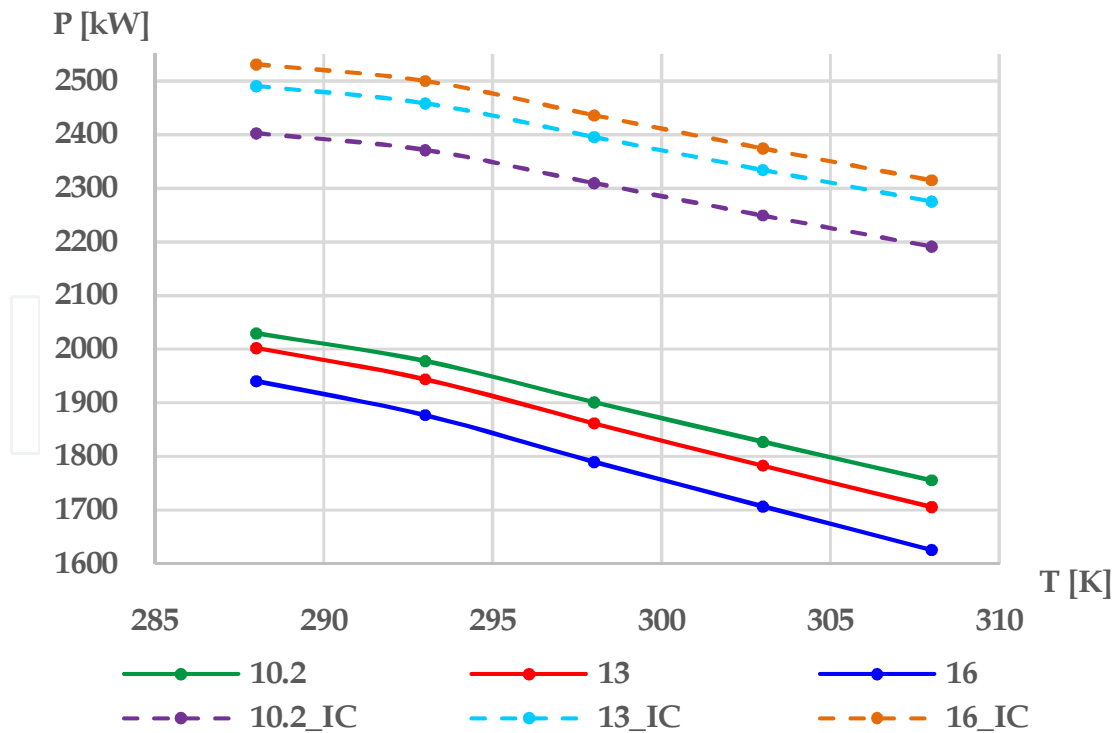


Figure 12. Gas turbine output power variation versus inlet air temperature (at the various compression ratio), for simple configuration and intercooling-IC

It is noticed a decrease of shaft power and thermal efficiency, simultaneous with an increase in specific fuel consumption, at an ambient temperature increase, phenomenon that occurs for all cases in which thermodynamic analysis was conducted, both in the initial gas turbine configuration, as well as in intercooling. For the initial configuration, a pronounced variation occurs with ambient temperature increase, easily noticeable in the case of high compression ratios. In the case of intercooling gas turbine, performance variation is approximately linear for all the three compression ratios, with a decrease in power of 2.5 % for each 5 degrees of the environment.

In the field of afterburning, the theoretical research has sought: to obtain a reduced pollutants afterburning installation, in comparison with the existing cogeneration power plant 2xST 18 (figure 3); afterburning installation flexibility when supplied with gaseous fuels; to study the influence of water spraying into the gas turbine's combustion chamber upon the afterburning [14, 15]. Thus, the numerical simulations performed in CFD environment have showed that (at nominal conditions), by modifying the afterburning module of the cogenerative plant 2xST 18, the NO_x concentration in the exhaust gases is lowered three times [14, 15]. The base module of the cogenerative plant has been mainly modifying by flaring at 15° and by introducing a concentrator.

5. Inlet air cooling installation and afterburning integration within gas turbine cogenerative group – Future research

Gas turbines usually function with high air excess, between 3 and 6 at nominal regime, and even higher at partial-load regimes [34]. This enables the use of an afterburning module, but the control of the air/fuel mixing requires the minimization of the air excess. Thus, reducing the exhaust emissions must be balanced against providing the cogenerative group's performances in a flexible manner.

5.1. Efficiency-emissions binom

The main pollutant emissions produced by gas turbines consist of: nitrogen oxides (NO_x), carbon monoxide (CO), volatile organic compounds (VOCs). Gas turbines typically operate at high loads which make their design for optimum combustion and maximum efficiency to be made at nominal load. Controlling the concentration of all pollutants is difficult in the conditions of variable loads operating. In high loads operating regimes the concentration of NO_x is higher, while in lower loads operating regimes (under 50 %) the thermic efficiency decreases and the concentrations of CO and volatile organic compounds increase. Thus the factors which determine the formation of pollutants in the exhaust gases are [14, 15]: the temperature and the excess of air in the primary zone; the process homogenization degree in the primary zone; the combustion process products residence time; the "freezing" characteristic of the reaction near the fire tube; etc. To decrease the emissions of NO_x is necessary to reduce the temperature in the area in which the combustion reaction takes place and in the high temperature zones, respectively to rethink the distribution of the air flows (combustion

in stages). The final version of the gas turbine's combustion chamber will be a compromise between the level of the pollutant emissions, performance and flexibility. Depending on the combustion temperature, in figures 13 and 14 are represented the NO_x and CO emissions levels typical for a class of industrial gas turbines, using different fuels and at various operating regimes [35]. The high CO emissions level indicates an incomplete combustion and a decrease in efficiency. From figures 13 and 14 it can be observed that for temperatures up to 760°C , the levels of NO_x and CO are comparable (especially for natural gases), while in the case of higher temperatures the NO_x emissions increases rapidly, while the CO emissions level remains practically constant. By comparison, in the case of micro gas turbines, operating at 70-100 % loads, the CO emissions are low (figure 15) but they increase fast when operating at under 70 % load [36]. In the case of micro gas turbines the NO_x emissions level are low over a wide range of operating regimes (30-100 % load). Adding a heat recovery steam generator and a cooling system for the intake air (with fog) could be a solution for increasing the performance and decreasing the influence of high temperature in the summer. The high content of vapours in the combustion gases (by injection of water/steam) leads to: acid corrosion (when using fuels which contain sulfur); increase thermal solicitations of the combustion chamber, reduce the heat recovery level, etc. The exhaust gases flow at the gas turbine exit is turbulent and uneven in the transversal section. Thus there might appear reverse flows in some areas of the heat recovery steam generator transversal section. The unevenness of the flow at the combustion chamber exit and the variation of the exhaust gases composition influence the functioning of the afterburning module. Thus the afterburning is influenced in terms of efficiency, pollutants, flame stability but also in terms of corrosion of the elements subject to the action of the exhaust gases. Generally, for a good design of the exhaust gases flow into the heat recovery steam generator, the following factors must be taken into account [15]: the geometry of the gas turbine exhaust and its direction; the size of the heat exchange surfaces; the position of the afterburning; mass flow and mean speed at the gas turbine exit; local speeds near the walls and at the entrance of the first heat exchange surface. In general the gas turbine's evacuation is not directly coupled with the heat recovery steam generator; after leaving the evacuation of the gas turbine (in the case of the cogenerative power plant 2xST 18 – Suplacu de Barcau, figure 2), the exhaust gases pass through a silencer, a by-pass assembly, an adaption section to the afterburning and then they reach the afterburning chamber. An uniform distribution of the flow in the transversal section insures a proper functioning of the heat recovery steam generator, especially of its overheater. This creates the necessary premises to ensure low emissions for the cogenerative group. If the exhaust gases coming from the turbine or the air flow are not evenly distributed, in the same way as the fuel, serious variations of the temperature can appear downstream of the burner.

In general, the variation of the speed (upstream of the burner), on 90 % of the burner section, mustn't exceed $\pm 15\%$ of the mean speed on the whole transversal section. In reality the exhaust gases temperature, downstream of the burner, will never be perfectly uniform. Even with a perfect distribution of the gas flow in the turbine, upstream of the burner, the temperature of the gases in the zone of each afterburning module will be higher than the temperature in the areas between the modules. These requirements must fall in the market trends, where (in terms of fluctuation of the electricity and fuel prices) the flexibility in functioning has become a major

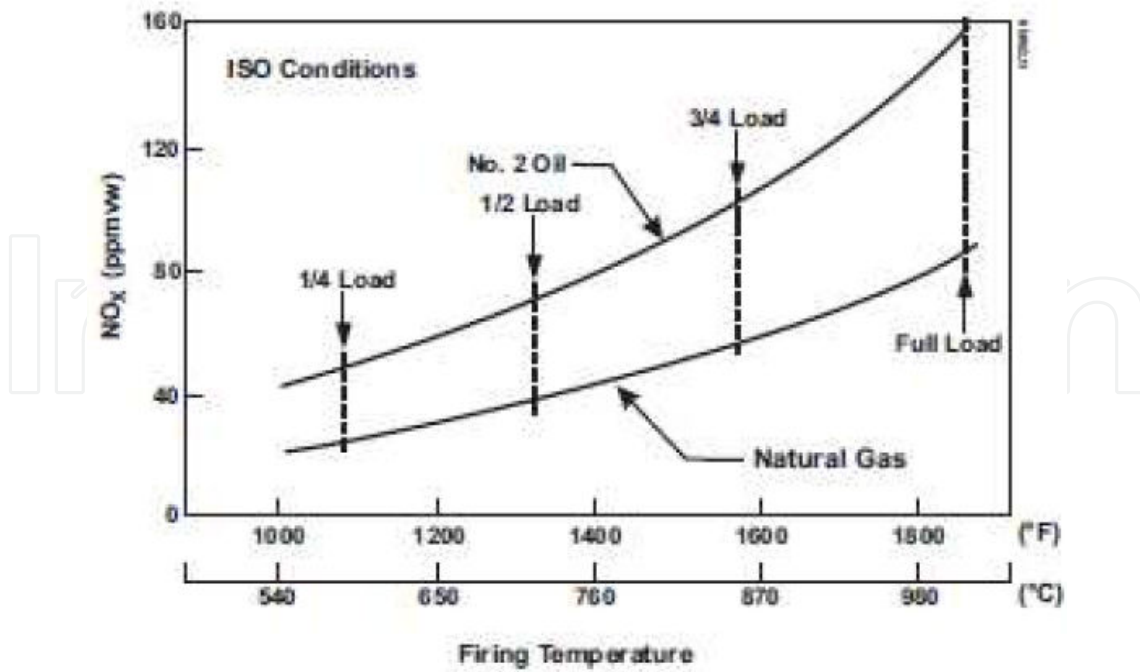


Figure 13. NO_x emissions variation for a class of industrial gas turbines - © 2010 Richard 'Layi Fagbenle. Originally published in [35] under CC BY-NC-SA 3.0 license. Available from: <http://dx.doi.org/10.5772/10206>

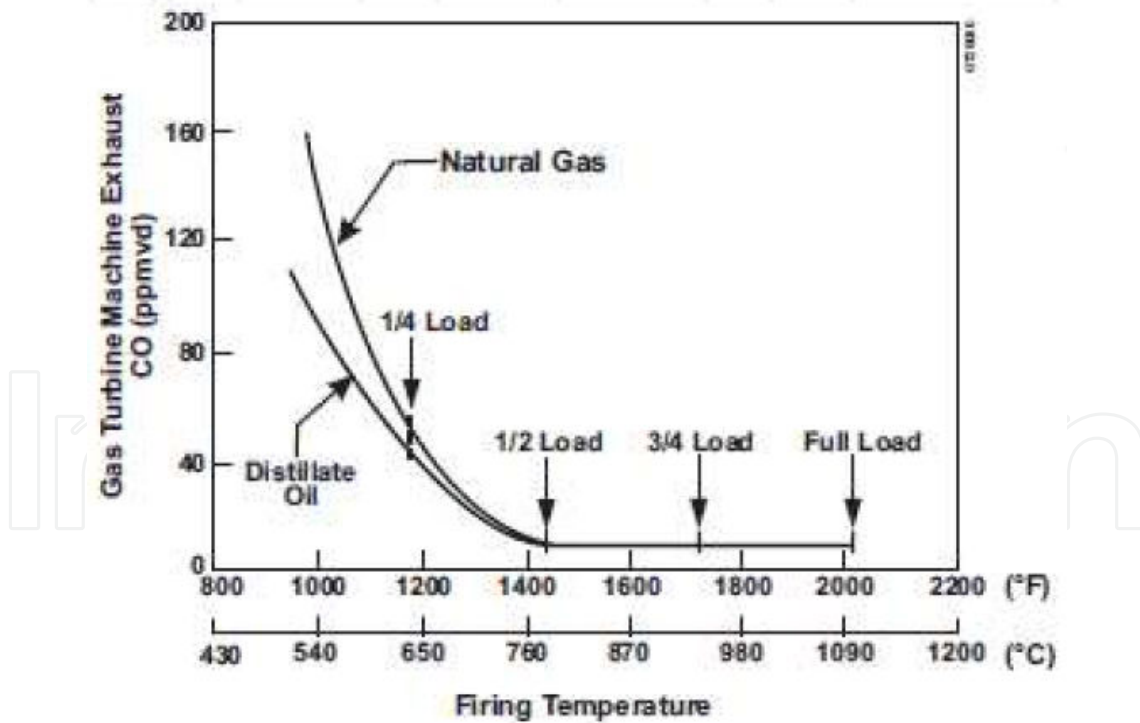


Figure 14. CO emissions variation for a class of industrial gas turbines - © 2010 Richard 'Layi Fagbenle. Originally published in [35] under CC BY-NC-SA 3.0 license. Available from: <http://dx.doi.org/10.5772/10206>

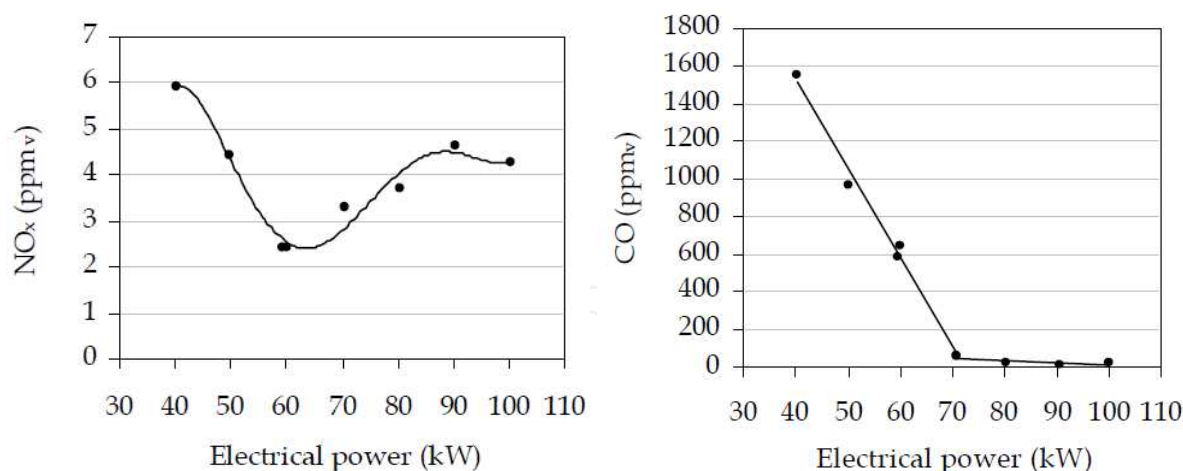


Figure 15. NO_x (left) and CO (right) emissions variation for a class of micro turbines - © 2010 Flavio Caresana, Gabriele Comodi, Leonardo Pelagalli, Sandro Vagni. Originally published in [36] under CC BY-NC-SA 3.0 license. Available from: <http://dx.doi.org/10.5772/10211>

subject, new concepts being imposed. Increased flexibility aims [37]: rapid start and stop; rapid change of the load; increase reliability in the case of quick start and load predictability; frequency control and of auxiliary services.

5.2. Experimental research conducted at INCDT COMOTI Bucharest

The experimental research carried out at INCDT COMOTI Bucharest, in close relation with the theoretical research (see chapter 4), have concentrated on: obtaining an afterburning installation on gaseous fuel, with low emissions; the realization of water spraying systems and their testing; experimenting water injection in the intake device of the gas turbine, at nominal functioning regime.

The numerical simulations carried out in CFD environment show that, by modifying the afterburning module of the cogenerative power plant 2xST 18, three times lower NO_x emissions can be obtained at nominal functioning regime [14, 15]. Up until now there have been carried out comparative experiments (figure 16) using gaseous fuel, at partial load (3 % of the nominal load). Thus the experiments at partial load show a reduction of the NO_x emissions with 30 %. From figure 16, right, it can be observed that the flame better fills the fire tube of the new afterburning module, and the temperature field is more uniform, being in good correlation with the results.

Before beginning the experiments regarding the effects of water injection in the intake device of the TA2 gas turbine, there has been studied the form of the water spray jet using a liquid fuel atomizer from TV2-117A gas turbine (figure 17). In these tests there has been chosen the atomizer circuit which ensures a droplet diameter (figure 17 left) comparable with the dimension of the gas turbine's intake device (figure 18 left). During the tests the water (untreated) has been sprayed at a pressure of 30 ± 0.5 bar, the atomizer being position on the TA2 gas turbine axis. The distance between the atomizer and the gas turbine intake has been varied between 1500 mm and 2000 mm. The testing rig is composed by: TA2 gas turbine,

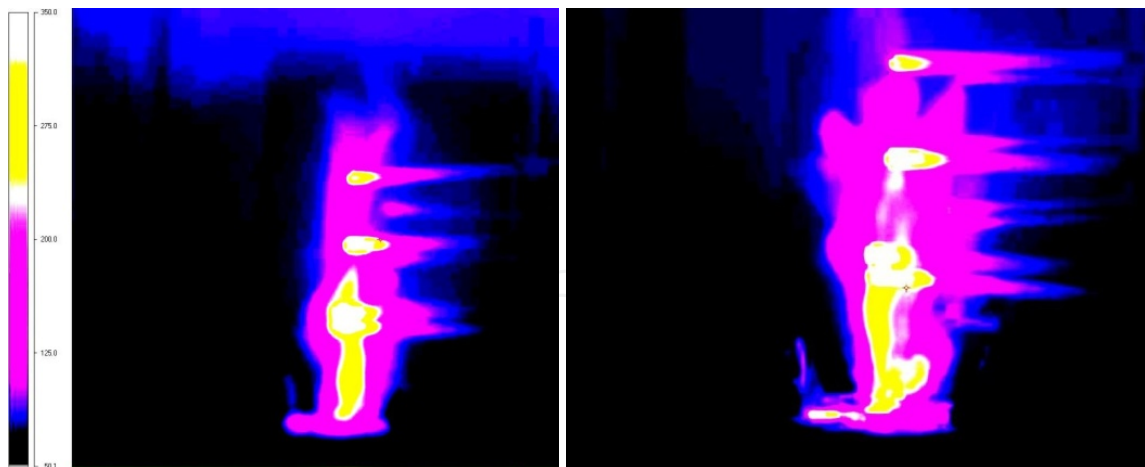


Figure 16. Comparative analysis, in infrared, of the afterburning modules (the existing module on the 2xST power plant – left; numerical simulation result – right) [15, 38]

natural gas afterburning installation – positioned on the vertical, water spraying installation, command and data acquisition chamber. The emissions measurement has been realized using the Horiba PG 250 gas analyzer, positioned on the stack (the exit of the afterburning chamber – figure 18 right). The experiments conducted on the TA2 gas turbine, using gaseous fuel, have been carried out at starting regime (10500 ± 6 rpm; 75 ± 5 Nm³/h natural gas flow).



Figure 17. Water spraying tests, using a liquid fuel atomizer from TV2-117A gas turbine

The mean temperature T_{3M} (the gases mean temperature before the gas turbine) has been determined using 17 thermocouples (figure 19 left), using the methodology presented in [15]. The free turbine speed has resulted from the vibration analysis, data acquisition being realized using module IEPE BNC Ni9233 (National Instruments). The exhaust gases temperature (T_5 – figure 19 right) has been measured using a thermocouple.

The experiments included several series of 20 minutes in which the gas turbine functioned at starting regime (without the afterburning module), using natural gases as fuel (for 10 minutes



Figure 18. Experiments regarding the injection of water in TA2 gas turbine (left) and gas analyser PG 250 (right)

the gas turbine functioned without water injection, followed by 10 minutes of functioning with water injection). The results regarding NO_x emissions during the experiments conducted on the TA2 gas turbine using natural gases as fuel, with/without water injection (the atomizer is positioned on the gas turbine's axis, at 1500 mm from the gas turbine's intake device) are presented in figure 20. The results obtained so far regarding water injection in the intake device of TA2 gas turbine (for power increase) are more qualitative, although by introducing water the speed of the free turbine is increased with about 20 rpm. Simultaneously with the introduction of water, temperatures T_{3M} and T_5 decrease with about 5°C , and the NO_x emissions decrease with about 4 ppm. It is expected that the influences will be more conclusive when the load of the gas turbine and the water quantity are increased.

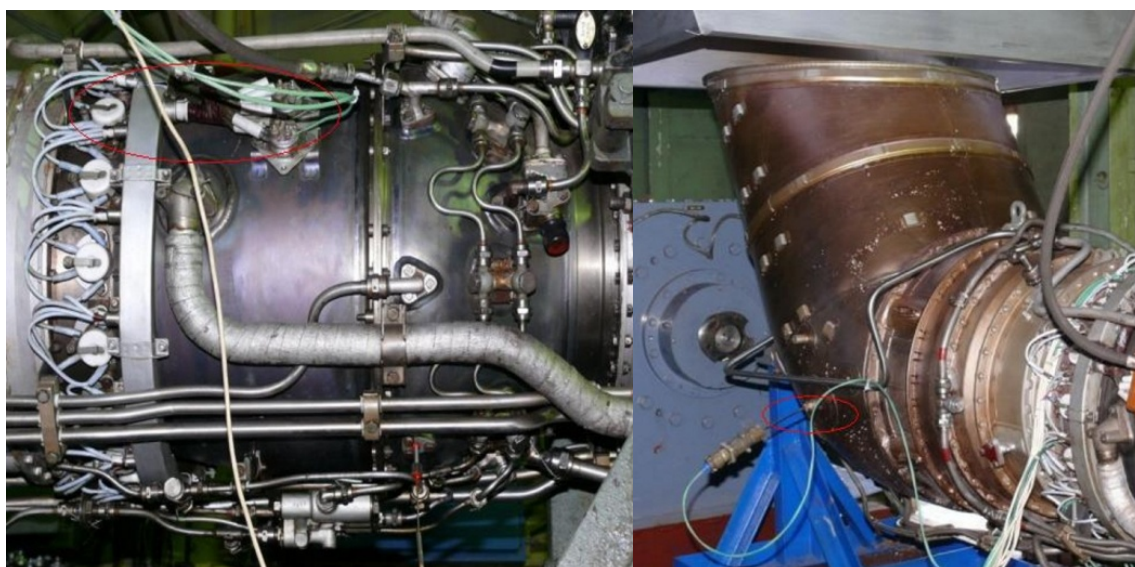


Figure 19. The thermocouples position for determining T_{3M} (left) and T_5 (exhaust device exit - right)

After about 10 series of experiments, of 20 minutes each, the boroscopic examination of the gas turbine didn't revile any significant deposits. Possibly in the future it will be necessary to appropriately treat the water injected in the intake device.

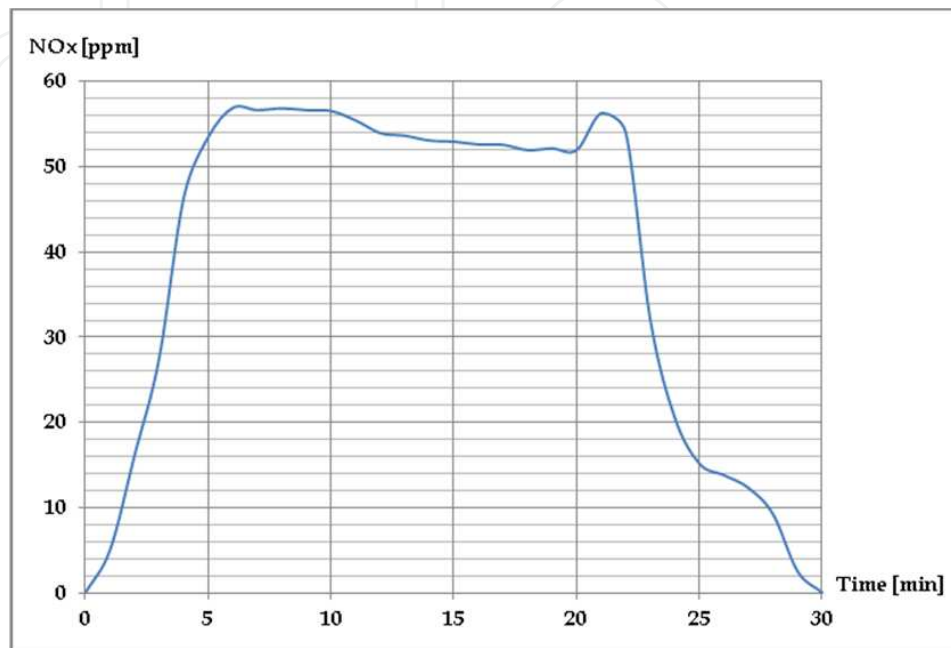


Figure 20. NO_x emissions variation during the experiments carried out on TA2 gas turbine, with water injection in the intake device

5.3. Future research

Future research to be conducted at INCDT COMOTI Bucharest will follow the general context given by the efficiency-emissions reduction-flexibility triad, by numerical simulations and experiments concerning: processes that take place during the cooling of the intake air; combustion in the gas turbine and afterburning installation; increase of the efficiency and pollutants reduction; flexibility regarding the used fuels.

6. Conclusions

1. The selection of a location for a gas turbine cogenerative plant imposes climatic conditions and demands adequate technical solutions to meet performance requirements, especially during summer season when inlet air temperature rises, leading to a decrease in power output and efficiency.

2. Water content modifies thermodynamic properties of intake air (density, specific heat) affecting power output and heat mass flow resulted from the gas turbine. If in the past air humidity was neglected, in present day cogenerative gas turbine power increase, but also water/steam injection impose the need for it to be taken into account.
3. The main research directions in the area of cogenerative groups with gas turbines efficiency increase are: combustion temperature increase; compression ratio increase; improvement of design methods, combustion technology and advanced materials; technological transfer for aviation domain to industrial turbines domain; integrated systems (combined cycles, intake air cooling, exhaust turbine gases heat recovery, afterburning, etc.).
4. Determinant factors concerning the overall efficiency of the cogenerative group are: gas turbine exit temperature, temperature at the heat recovery steam generator stack; ambient environment temperature. For these the most influential factor upon the increase of the overall efficiency is the temperature at the heat recovery steam generator stack.
5. Operating flexibility of equipment has become a major subject. Gas turbines are designed to function generally at nominal regime, in maximum efficiency conditions and minimum pollutants. At cogenerative groups with heat recovery steam generator, for producing technological steam, is preferable that the flexibility to the process demands to be achieved using afterburning installation.
6. Theoretical and experimental research conducted at INCDT COMOTI Bucharest, allowed: to be showed that, in the case of a gas turbine with intercooling, the performances variation is approximately linear for a compression ratio between 10.2-16, with a power decrease of 2.5 % for each 5 degrees increase of the environment temperature; to be obtained a afterburning module with a 30 % reduction of the NO_x reduction (at partial load) in comparison with the existing cogenerative power plant 2xST 18 – Suplacu de Barcau; to demonstrate the power increase and NO_x emissions reduction when injecting water in the intake device of TA2 gas turbine.

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