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# INFLUENCE OF AMBIENT CONDITIONS ON AN AERODERIVATIVE GAS TURBINE BASED COGENERATION PLANT - A COMPARISON OF NUMERICAL SIMULATION WITH FIELD PERFORMANCE DATA

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## ABSTRACT

Gas turbine performances are directly related to outside conditions. The use of gas turbines in combined gas-steam power plants, also applied to cogeneration, increases performance dependence by outside conditions, because plants boundary conditions become more complex.

In recent years, inlet air cooling systems have been introduced to control air temperature and humidity at compressor inlet resulting in an increase in plant power and efficiency.

In this paper, the dependence of outside conditions for an existing cogenerative plant, located in Tuscany (Italy), is studied. The plant is equipped with two GE-LM6000 aeroderivative gas-turbines coupled with a three pressure level heat recovery steam generator, cogenerative application being related to the industrial district. The ambient temperature has been found to be the most important factor affecting the plant performance, but relative air humidity variation also has considerable effects.

The field performance data are compared with a numerical simulation. The simulation results show a good agreement with the field performance data. The simulation allows evaluation of design and off-design plant performance and can become a useful tool to study the outside condition influence on power plant performance.

## NOMENCLATURE

$T$ = Temperature [°C]
$Tr$ = Torque [kN·m]
$x$ = Steam mass fraction related to dry-air quantity $[kg_{steam}/kg_{dry\text{-}air}]$
$W$ = Net power output of gas turbine [MW]
$\beta$ = Compression ratio
$\phi$ = Relative humidity
$\eta$ = Efficiency
$\omega$ = Rotation speed [rad/s]

### Subscripts and superscripts

inlet = compressor inlet
out = outside air
max = maximum value: firing temperature
HP = high pressure shaft
LP = low-pressure shaft: from compressor to turbine shaft
PS = power shaft

## INTRODUCTION

Aeroderivative gas turbines compared to heavy duty gas turbines, are more efficient and compact because higher compressor ratio and firing temperature are used. The aeroderative gas turbines present a good part load performance, but the introduction of DLN combustion determine some problems in this field (Casper, 1993).

Generally, the gas turbines and are strongly affected by ambient conditions that influence output power. The gas turbine shaft configuration and load condition influence these effects. For example, studying a single shaft gas turbine for electric power generation (frequency is constant), when the ambient temperature increases, the air density decreases, so the inlet air mass flow rate of the compressor decreases; imposing a constant firing temperature, the turbine requires a lower expansion ratio to equilibrate the reduction of compressor mass flow rate. Using compressor and turbine characteristic curves, a new operative condition can be determined: the compressor goes

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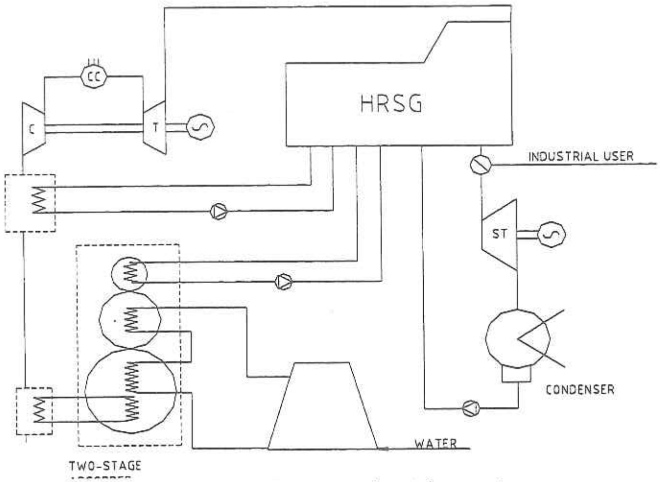

**FIGURE 1: Power plant layout.**

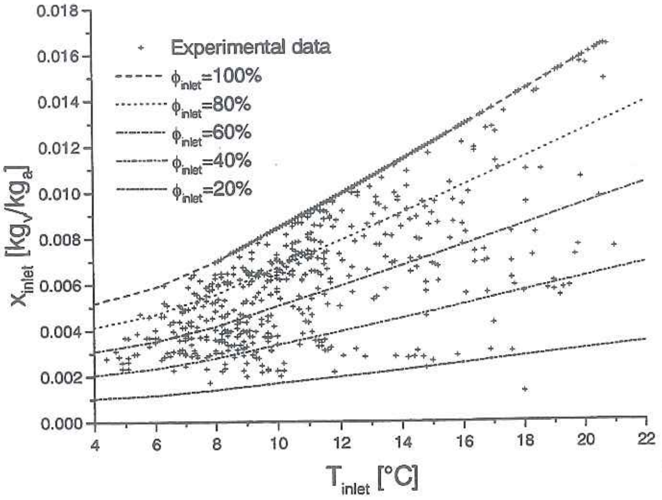

**FIGURE 2: Scatter of experimental data.**

toward chocking where compressor efficiency is lower. The behaviour of multi-shaft gas turbine is similar but less relevant. Furthermore, when inlet air temperature increases, the specific work of the compressor also increases (in accordance with thermodynamic analysis).

Inlet air cooling systems (IAC systems) have been designed or studied to enhance the gas turbine performances. These systems cool air at the compressor inlet and keep the gas turbine nearer maximum power and efficiency conditions.

Air cooling through humidification obtained from spray nozzles at the compressor inlet is the cheapest way. The evaporation inside the compressor reduces the air temperature, so specific compression work decreases, but water drops inside rotor vanes may cause early blading wear. Humidification is very low when outside temperature is near saturation (Utamura et al., 1999).

More complex systems require absorption chillers or compression refrigeration units. Absorption chillers are more expensive than compression cooling systems, but they use recovery thermal energy as a primary power source and are found more reliable (De Lucia et al., 1993; De Lucia et al., 1995). Furthermore, compression-cooling systems have a performances drop at variable loads and should, therefore, be integrated with thermal energy storage systems. The choice of such plants must be based on technical and economic considerations (Ebeling et al., 1992; Ebeling et al.,1994).

Generally, previous works have studied air temperature effects on turbine performances, but all the control systems of the air temperature are also related to humidity which has side effects for turbine performance.

The aim of the present work is to quantify and explain the gas turbine performance of the GE-LM6000 aeroderivative gas turbines installed in a cogenerative plant located in Tuscany (Italy). The dependence of outside conditions of a real cogenerative plant is shown by means of an analysis of plant parameters collected by sampling on a yearly basis. A numerical simulation with a code has been realised in order to better understand the experimental results.

## THE COGENERATIVE PLANT

The combined plant meets the paper industry steam requirement and, it allows a maximum power production of 100.MW.

Figure 1 shows the plant layout schematically. The plant is equipped with two GE LM6000 gas turbines which burn natural gas with DLN combustion chambers.

The turbine exhaust gases are directed to a three-pressure level fired heat recovery steam generator that produces steam at 65 bar, 6.5 bar and 2.5 bar. The maximum mass flow of the steam produced by each level is 20.28 kg/s, 1.90 kg/s and 1.81 kg/s, respectively. Firing of HRSG could be necessary to meet process steam peak requirement. The third pressure level is not equipped with superheater shells and provides alternatively steam to an anti-ice panel and a two stage chiller for the air conditioning at the compressor inlet. The chiller introduction is very important to increase performance when the ambient temperature is high.

The distributed control system of the plant collects data, on an hourly basis, and turbine power output, outside and inlet air temperature and outside air humidity. Inlet air humidity is determined by the previous data, since air cooling takes place initially along a constant steam mass fraction curve, and successively along the saturation curve up to the final air temperature.

Figure 2 shows the distribution of experimental data on the temperature versus humidity plane which is to be taken into account to identify the domain areas where interpolations offer a better confidence level.

Figures 3 and 4 show the statistical distribution of outside parameters collected by sampling on a yearly basis: for each month, three days were been chosen at random and the respective data considered. Continuous curve on the graph represent a Gaussian curve which fits the data distribution. All temperatures in figure 3 are over 5°C because the anti-ice panel prevents ice formation at the compressor inlet. Figures 5 and 6 show statistical distributions for air characteristics at inlet to the compressor.

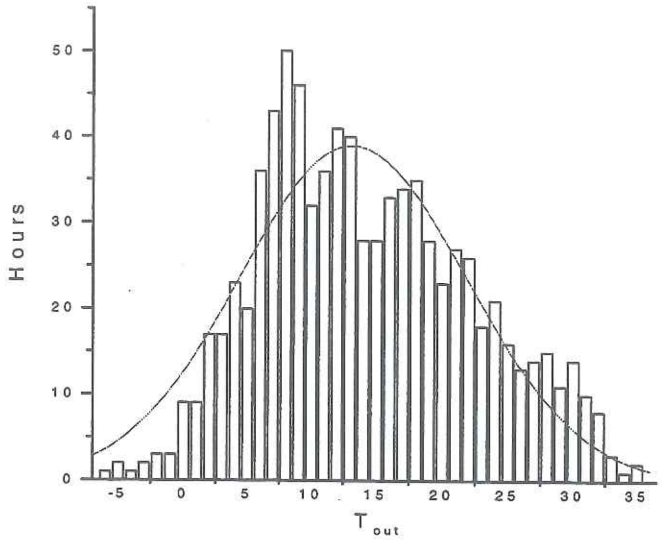

FIGURE 3: Statistical distribution of outside air temperature.

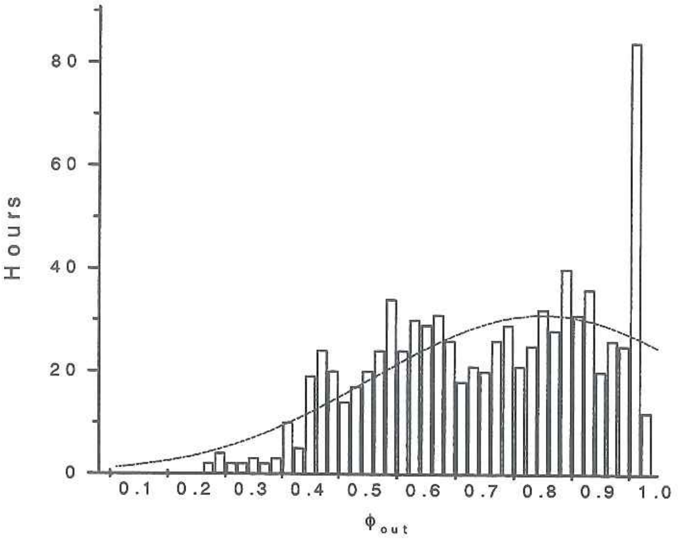

FIGURE 4: Statistical distribution of outside (ambient) relative humidity.

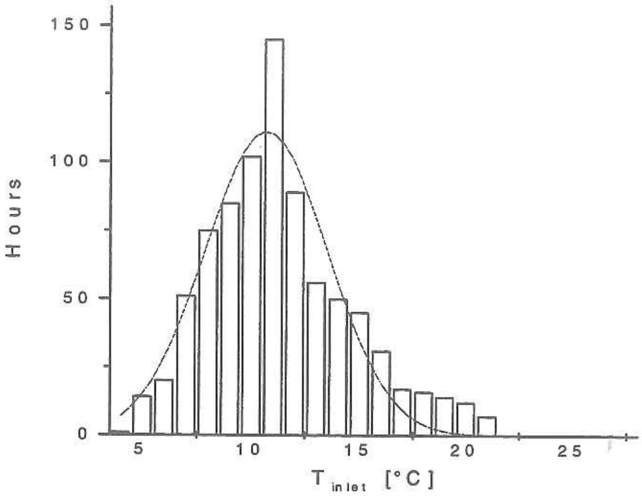

FIGURE 5: Statistical distribution of inlet air temperature.

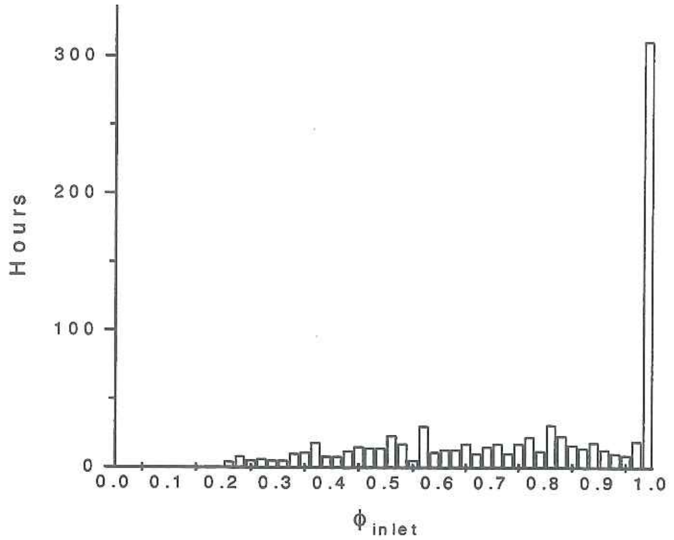

FIGURE 6: Statistical distribution of inlet air humidity.

It can be noted that the air conditioning system reduces temperature scattering to an average value of 10°C, keeping the gas turbine nearer to the maximum power condition.

As the IAC system decreases the air temperature, it also increases relative air humidity, so that the air at the compressor inlet is mainly near saturation (Figure 6).

In order to find a relation between outside conditions and gas turbine performances, inlet air temperature and humidity were correlated to the developed power by means of a two-variable interpolation which showed turbine performances -outside dependence. The results of interpolation are shown in figure 7. The most significant variable, which affects output power, is the air temperature, but the surface also has bending in the humidity direction.

When the air temperature is high, the humidity effect on gas turbine performance is greater because the air can reach higher values of steam mixed with dry air. Figure 7 shows that inlet air dehumidification cannot be considered as a way of increasing power output. This thermodynamic transformation cannot be realised at a constant temperature, since it requires air cooling along the saturation curve, followed by air heating at constant steam mass fraction up to the required relative humidity. Air heating always takes place following the surface in the direction of power decrease.

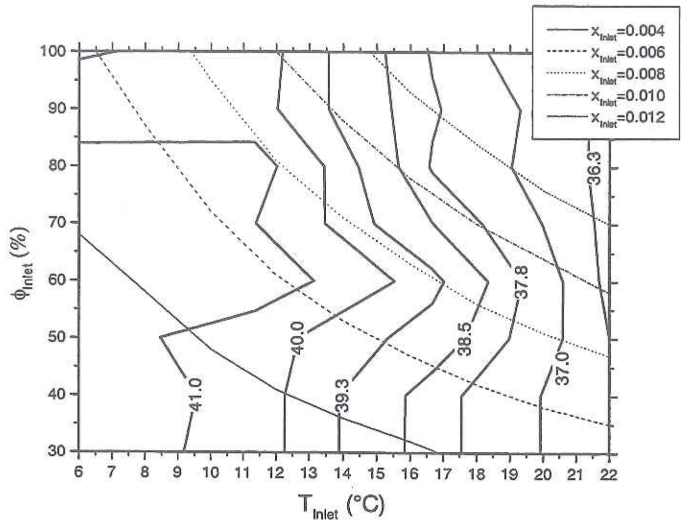

**FIGURE 7: Results of interpolation, gas turbine performance - outside dependence.**

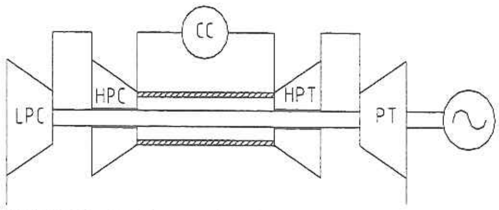

**FIGURE 8: Schematic layout of GE LM6000.**

Dehumidification, in the end, involves the variation of a more important parameter (the temperature) in such a way that it is negative for output power production.

Humidity does not affect gas turbine performance to a great extent, but its influence is unavoidable.

A numerical simulation of GE LM6000 gas turbine with a code developed at the University of Florence (Carcasci and Facchini, 1996), was carried out to better explain the influence of ambient conditions.

## MODULAR CODE

The proposed studies require the use of adequate calculation tools for power plant simulation and performance predictions, particularly to analyze off design performance. The authors have used the modular code already developed by Carcasci and Facchini (1996); the reader is referred to previous papers (Carcasci and Facchini, 1996; Carcasci et al., 1996; Carcasci et al., 1997) for a complete presentation of the modular approach used.

A modular simulation code must be able to create a new power plant configuration, without creating a new source program. The code must also be able to handle any combination of input data. If it is necessary, the modular code easily allows addition of new components (Carcasci and Facchini, 1996).

The power plant configuration is defined by connecting a number of elementary components representing different unit operations such as compressors, pumps, combustion chambers, splitters, mixers, etc. Thus each component is defined as a black box capable of simulating a given chemical and thermodynamic transformation. All equations defining the power plant are linearized (the coefficients are however updated in the course of the calculation), then all equations are solved simultaneously using a classic matrix method. With this approach, none of the data describing the different components of the system are considered essential, unlike a number of other semi-parallel or sequential methods.

## NUMERICAL SIMULATION

The method used in this study allows simplified component simulation resulting in a better description of cycle behaviour and a better understanding of manufacturer provided information.

For off-design performance evaluation, the unit description becomes more complex and requires a more detailed design approach.

### Design analysis

Figure 8 schematically shows a layout of GE LM6000 gas turbine.

The design study requires a geometric description of the component (Facchini, 1993; Carcasci et al. 1996), which allows identification of the characteristic parameters which can then be used in typical off-design correlations (e.g., the velocity triangle at mean radius and other cascade parameters for the compressor or turbine). For this parameter identification, the knowledge of some of the manufacturer's data can be important to improve simulation results.

Standard air conditions at the compressor inlet have been assumed: inlet air temperature is fixed at 15°C and relative air humidity at 60%. Furthermore, some gas turbine data, like nominal output power and efficiency, are well known from the GE catalogue, but the other geometric data, like deflection angles of the flow, have been assumed.

The code determines other parameters like mass flow, pressure and temperature at each thermodynamic point of the cycle. These data are necessary to define the gas turbine operating conditions completely.

### Off design performance analysis: inlet air temperature variations at constant relative humidity.

Gas turbine behaviour was simulated at working conditions that differ from nominal. The off-design study is based on fixed geometry (obtained by the design study), and there is a reduction in the number of input data.

An initial off design simulation was realised at variable inlet air temperature and constant relative humidity ($\phi_{Inlet}$=60%).

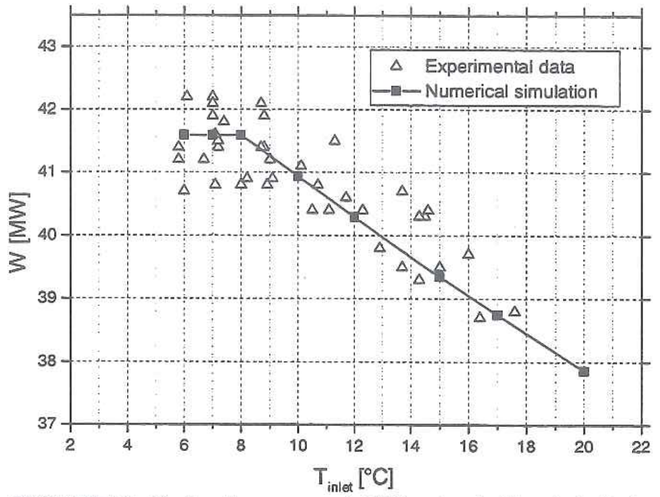

FIGURE 9: Output power at 60% of relative inlet air humidity and variable inlet air temperature.

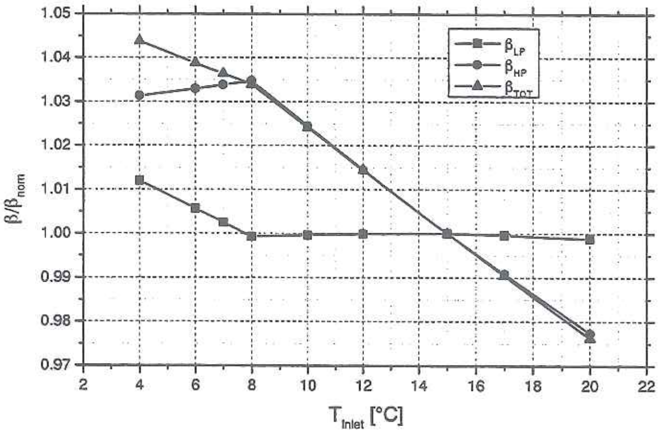

FIGURE 10: Compression ratios at variable inlet temperature and 60% relative inlet air humidity.

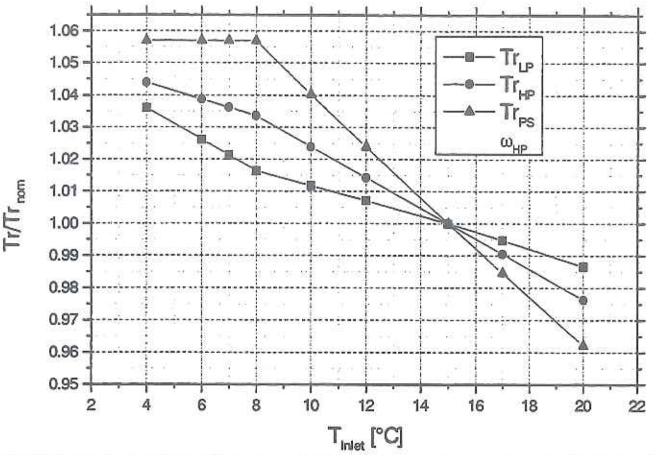

FIGURE 11: Relative torque values at variable inlet air temperature and 60% relative inlet air humidity.

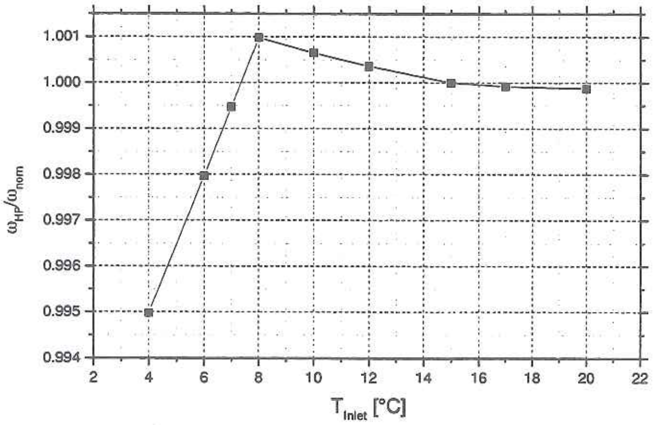

FIGURE 12: High-pressure shaft rotation speed at variable inlet air temperature and 60% relative inlet air humidity.

Figure 9 shows gas turbine output power as a function of ambient temperature obtained using simulation. The result of numerical simulation shows a good agreement with the field data. When air temperature at the compressor inlet is lower than 8°C, there is a limitation of output power. To impose this limit a control system is used; in the present work, Fuel Control (decreasing of the maximum temperature of the cycle) is used. The maximum temperature decreases a little, so this reduction can be compatible with DLN combustion chambers.

In order to understand what critical parameter causes control system involvement, variations of compression ratios, torque on each of the three shafts and high pressure shaft rotational speed are shown in figures 10, 11 and 12, respectively. All data on these graphs are related to respective design values. When the inlet temperature decreases, the air density is greater and the values of all torques increase. Inlet air temperature reduction does not particularly affect rotational speed of high-pressure shaft. Torque on the power shaft grows more than others: it is proportional to output power because the rotation speed of power shaft is fixed. Probably, when air temperature is 8°C, power shaft torque reaches the maximum value.

It can be noted (Fig. 12) that, with certain inlet air characteristics, the fuel control system, decreasing $T_{max}$, causes a deceleration of the high-pressure shaft, reducing the torque on this shaft less than it influences torque on power shaft. In this case, the presence of LP compressor at constant rotational speed determines a progressive increase of mass flow rate at the compressor inlet. The high-pressure compressor work slightly increases and relative torque similarly increases, considering speed reduction of high pressure turbine. In the other hand, when the fuel control system is activated the pressure at the high pressure turbine exit becomes greater (figure 10) and temperature decreases. Globally, the gross work of the low pressure turbine increases because the air mass flow rate increases, but this work growth is totally used up by the low pressure compressor and so

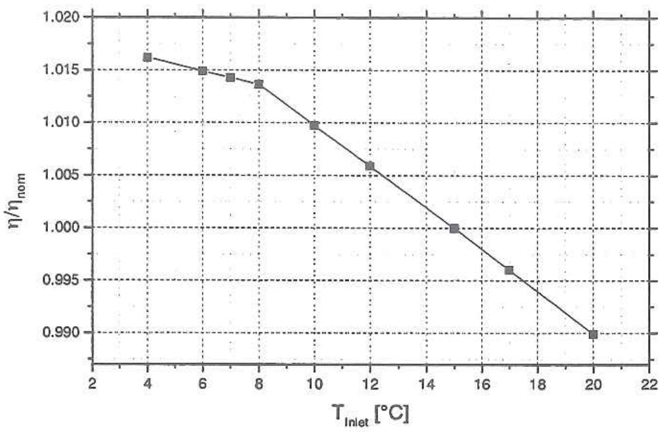

FIGURE 13: Relative efficiency at variable inlet air temperature and 60% relative inlet air humidity.

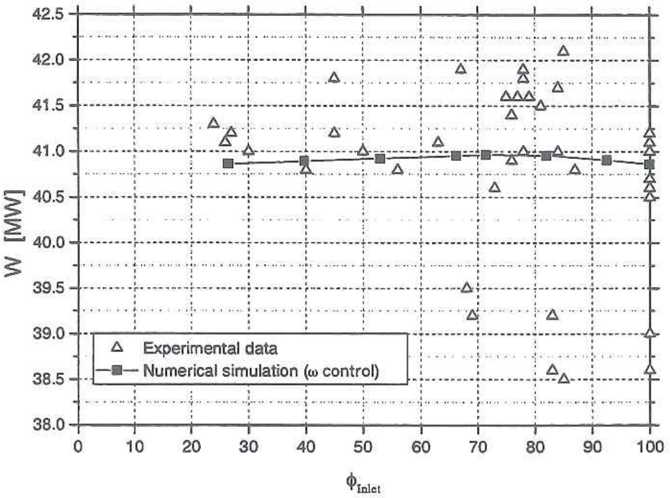

FIGURE 14: Net output power at variable air inlet humidity and 10°C of inlet air temperature.

the required constancy in net power output is achieved. Furthermore, control system involvement causes an increase in torque on the low pressure compressor shaft. Since the compressor is constrained to rotate at constant speed, its working conditions move to the stall condition. Thus, the critical parameter that involves control system intervention seems to be the torque on power shaft.

The variations of gas turbine efficiency at variable temperature are shown in figure 13.

### Off design performance analysis: inlet air humidity variations at constant ambient temperature.

In order to evaluate the humidity influence a simulation was carried out, keeping $T_{inlet}$ at constant value of 10°C, as near this value the experimental data scattering is lower. In figure 14, the simulation results are compared to the field performance data.

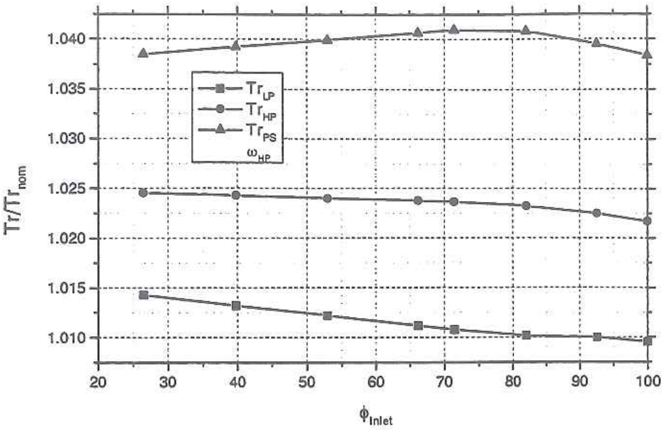

FIGURE 15: relative torque values at variable air inlet humidity and 10°C of inlet air temperature.

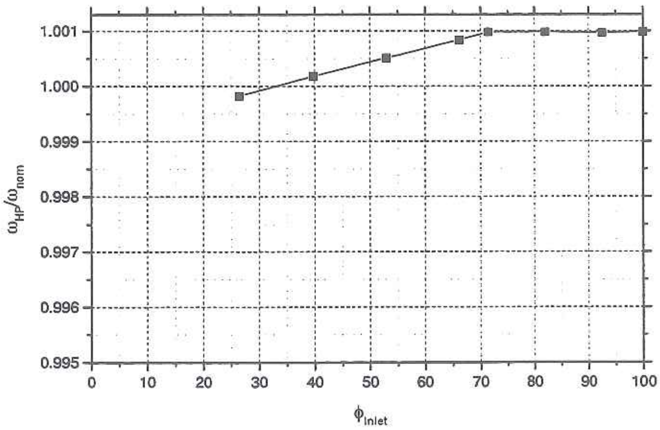

FIGURE 16: Relative high-pressure shaft speed at variable air inlet humidity and 10°C of inlet air temperature.

The small power variations due to humidity cause a wider data scattering and some of the data in figure 14 are out of range. These points are perhaps affected by random errors, but clogging of air filters at the compressor inlet has often been found when air humidity was high. Therefore, it is possible that the consequent losses in total pressure caused a reduction in net power output.

Figures 15 and 16 show the relative variations of torque on the shafts and high-pressure shaft' speed at different values of inlet relative air humidity.

Comparing figure 11 with figure 15, it can be noted that the torque on each shaft does not exceed the respective maximum value. A growing amount of water vapour in the air leads to a decrease in density and to an increase in constant pressure specific heat. Consequently the low-pressure compressor mass flow decreases; by increasing the enthalpy stage of the turbine the result is a slight increase in the net power and a slight acceleration of the high-pressure shaft.

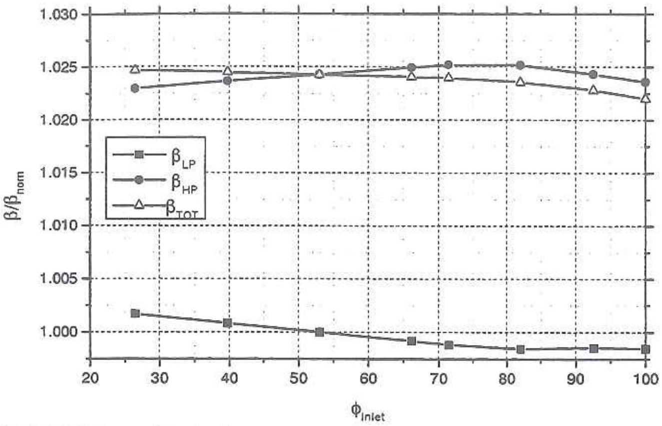

**FIGURE 17: Relative compression ratios at variable air inlet humidity and 10°C of inlet air temperature.**

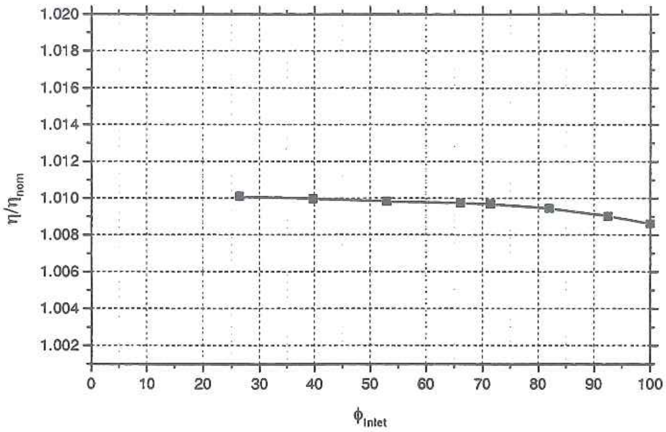

**FIGURE 18: Relative efficiency at variable air inlet humidity and 10°C of inlet air temperature.**

When $\phi_{Inlet} \cong 70\%$, the high-pressure shaft reaches the maximum speed value which was found with the previous simulation (figure 12, when $T_{inlet=}8°C$). Since interpolation of field performance data have highlighted a maximum in power output at variable relative humidity, it is possible that this gas turbine has the $\omega_{HP}$ nominal value near to the maximum value. In this case, even small variations of speed could not been tolerated, so the maximum in net power output may be caused by the involvement of the control system which limits high-pressure shaft speed. As in the previous case, it was supposed that the control system operates by small reductions of the maximum temperature of the cycle, but, at this time, the parameter which must be controlled is the high-pressure shaft speed. This leads to a slight decrease in net power when relative humidity exceeds 70% c.

In figures 17 and 18, the variations of compression ratios and efficiency of the gas turbine are shown. It can be noted that relative humidity at inlet does not greatly affect them.

## CONCLUSIONS

The influence of outside air temperature and humidity on a GE LM6000 gas turbine performances was shown. The most significant parameter is temperature, whereas humidity has less important side effects.

The effects of outside conditions on a twin-shaft turbine must be evaluated not only from a thermodynamic point of view, but also considering the interaction with the control system whose involvement is decisive for gas turbine response.

The use of numerical codes allowed simulation of gas turbine behaviour even with small variations in the working parameters, offering a good agreement with field performance data.

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## REFERENCES

Carcasci, C., Facchini, B., 1996; "A Numerical Method for Power Plant Simulations", *ASME Jnl of Energy Resources Technology*, March 1996, vol. 118, pp. 36-43. See also *IGTI Turbo Expo '95*, Huston (TX), June 2-6, 1995, 95-GT-269.

Carcasci, C., Facchini, B., Marra, R., 1996; "Modular Approach to Off-Design Gas Turbine Simulation: New Prospects for Reheat Applications", *ASME 1996 Turbo-Expo Symposium*, Birmingham (UK), June 1996, paper 96-GT-395.

Carcasci, C., Facchini, B., Harvey, S., 1997; «Modular Approach to Analysis of Chemically Recuperated Gas Turbine Cycles», *Flowers '97*, Florence (Italy), July 1997.

Casper R.L., 1993, "Application of the LM6000 for power generation and cogeneration", ASME paper 93-GT-278, presented at International Gas Turbine and Aeroengine Congress & Exposition, Cincinnati, Ohio – May 24-27, 1993.

De Lucia M., Bronconi R, Carnevale E., 1993, "Performance and economic enhancement of cogeneration gas turbines through compressor inlet air cooling", ASME paper 93-GT-71, presented at International Gas Turbine and Aeroengine Congress & Exposition, Cincinnati, Ohio – May 24-27, 1993.

De Lucia M., Lanfranchi C., Boggio V., 1995, "Benefits of compressor inlet air cooling for gas turbines cogeneration plants", ASME paper 95-GT-311, presented at International Gas Turbine and Aeroengine Congress & Exposition, Houston, Texas – June 5-8, 1995.

Ebeling J., Balsbaugh R., Blanchard S, Beaty L., 1994, "Thermal energy storage and inlet air cooling for combined cycle", ASME paper 94-GT-310, presented at International Gas Turbine and Aeroengine Congress & Exposition, The Hague, Netherlands – June 13-16, 1994.

Ebeling J., Halil Rick, Bantam D., Bakenhus B., Schreiber H., Wendland R., 1992, "Peaking gas turbine capacity enhancement using ice storage for compressor inlet air cooling" , ASME paper, presented at International Gas Turbine and Aeroengine Congress & Exposition, Cologne, Germany – June 1-4, 1992.

Erbes, M.R., Gay, R.R., Cohn, A., 1989; "GATE: A Simulation Code for Analysis of Gas-Turbine Power Plants", *ASME Turbo Expo '89*, June 4-8, 1989; Toronto, Ontario, Canada.

Facchini, B., 1993; "A Simplified Approach to Off-Design Performance Evaluation of Single Shaft Heavy Duty Gas Turbines", *ASME Cogen Turbo '93*, Bournemouth, UK, Sept. 1993.

Perz, E.W., Riesel, U., Schinagl, H.A., 1995; "A New Approach for Modelling Energy Systems", *ASME Cogen-Turbo Power Conference '95*, Vienna, Austria, August 23-25, 1995, 95-CTP-82.

Utamura M, Takaaki K., Hidetaro M, Nobuyuki H., 1999, "Effects of intensive evaporative cooling on performance characteristics of land-based gas turbine".