A MODEL OF COGENERATION PLANTS BASED ON SMALL-SIZE GAS TURBINES

S. Banetta, M. Ippolito*, D. Poli, A. Possenti
University of Pisa, Italy. * University of Palermo, Italy

A dynamic mathematical model of a generic cogeneration plant with microturbine is described and the corresponding regulating system is realized. The chief aim of this model is the evaluation of the services such a system is able to provide to an interconnected electric network, both in normal operation and in case of wide perturbations, or to an islanded grid after the separation from the main network.

MICROTURBINES FOR DIFFUSED COGENERATION

The electricity market liberalization is supplying new impulses to the competition between generation systems. Small dispersed generation in parallel to the main grid can represent a very interesting solution that reduces power flows and losses on the lines and contributes to the local voltage regulation. On the other hand, these small plants have a competitive efficiency especially when associated with thermal energy production (cogeneration). This is the case, for example, of densely-populated areas where electricity request is locally associated with heat demand, both for winter heating and for summer conditioning, also considering that heat production must necessarily be local.

In particular, the choice of microturbines for dispersed cogeneration is justified by low installation and maintenance costs, recent improvements in electric efficiency (up to 30%), high flexibility as regards the power sharing between thermal side and electrical one: that essentially means a payback time reduced to 3-4 years both in case of industrial and aggregated tertiary load.

The participation to the frequency regulation in case of large perturbation occurred on the main grid, as well as the ability to operate in a separated network and eventually to restart the main grid in case of its black out, could be very important "ancillary services".

In order to analyze the real possibilities of providing these services, for which the dynamic point of view and the running flexibility are fundamental, a "physical" model has been developed with Simulink, in order to define the general features of these systems, identifying their main limitations and performances.

THE PROCESS

The plant presents a gas microturbine and a high frequency synchronous machine, connected to the electric grid by a rectifier-inverter system. At the exit of the turbine, a bypass modulation valve allows the sharing of gas between an air-gas recuperator (to increase the electric power) and a boiler, which feeds a thermal user.

THE MODEL

The described model is non-linear, because of the adiabatic compression-expansion law for perfect gases into the turbine and the compressor, and evaluates the steady state and dynamic behavior of each component, without geometric details, simply respecting general physical laws. The realization peculiarities are transferred to the values of the heat exchange coefficients and to the fluid-dynamic resistances; the generality of this approach allows the use of this model also with plants having very different physical features and sizes and in very irregular operating situations for a model, like an out-of-service or a start up.

RESULTS OF SIMULATIONS

The steady state simulations carried out with the model show that the current technology makes this kind of plants suitable for dispersed cogeneration when the electric/thermal power ratio is in the 2-5 range.

The dynamic simulations demonstrate the very prompt response of the mechanical power provided at the shaft, obtained by modulating the fuel flow, so step variations of the required electric power are allowed without unacceptable perturbations to the main operating variables.

CONCLUSIONS

The described model allows the valuation of the solutions offered by co-generation systems with microturbines, in order to identify which problems can affect this kind of plants and design the regulating and protection system.

The simulations carried out with this model show that the problem concerning the supply of heat to the local users can be temporarily separated from the quick and large electric power variations required to support the main grid in case of important perturbations, or the local islanded grid in case of large out-of-services with separation from the main network; thanks to the current technology, these power demands can be met without exceeding the limitations posed by the constructor as regards to the rotation speed and the gas temperatures.

These considerations, together with the quick start of turbines and the prompt power and voltage modulation allowed by inverters, make systems with microturbine very useful to contribute to the different needs of an electric grid.
A MODEL OF COGENERATION PLANTS BASED ON SMALL-SIZE GAS TURBINES

S. Banetta, M. Ippolito*, D. Poli, A. Possenti
University of Pisa, Italy. * University of Palermo, Italy

ABSTRACT
In the following a dynamic mathematical model of a generic cogeneration plant with microturbine is described and the corresponding regulating system is realized.

The chief aim of this model is the evaluation of the services such a system is able to provide to an interconnected electric network, both in normal operation and in case of wide perturbations, or to an islanded grid in consequence of the separation from the main network.

The design of the regulating system takes into account the maximum acceptable power variations, in observance of the limitations the constructor has imposed to the process variables and respecting the requirements of the local thermal loads; the result is a quick and robust regulation, even in case of internal failures.

INTRODUCTION
The electricity market liberalization is supplying new impulses to the competition between generation systems. In this context small dispersed generation in parallel to the main grid can represent a very interesting solution that reduces power flows and losses on the lines and contributes to the local voltage regulation. On the other hand, these small plants have a competitive efficiency especially when associated with thermal energy production (cogeneration). This is the case, for example, of densely-populated areas where electricity request is locally associated with heat demand, both for winter heating and for summer conditioning, also considering that heat production must necessary be local.

In particular, the choice of microturbines for dispersed cogeneration is justified by low installation and maintenance costs, recent improvements in electric efficiency (up to 30%), high flexibility as regards the power sharing between thermal side and electrical one; that essentially means a payback time reduced to 3-4 years both in case of industrial and aggregated tertiary load.

The employ of clusters of turbines could represent another interesting development, since in case of islanded operation the use of a single turbine is often not acceptable.

Besides the above mentioned advantages provided by dispersed generation to the system management in case of parallel operation, the participation to the frequency regulation in case of large perturbation occurred on the main grid, as well as the ability to operate in a separated network and eventually to restart the main grid in case of its black out, could be very important “ancillary services”.

In order to analyze the real possibilities of providing these services, for which the dynamic point of view and the running flexibility are fundamental, the following model has been developed, not in order to simulate the behavior of a particular plant, but to define the general features of these systems, identifying their main limitations and performances.

THE PROCESS
The scheme of the plant is shown in Fig.1.

Figure 1 – Scheme of the analyzed process

Such a system, with a 45 kWe microturbine, is going to be installed at the University of Pisa by the Department of Electrical Systems and Automation and the Department of Aerospace Engineering, in order to partially meet the electric and thermal load of the Faculty of Engineering; this plant will be employed also for the identification tests of the model and for the performance verifies.

THE MODEL
A “deep” model, based on energy and mass balances and on general operating principles of compressors and gas turbines, has been developed with Simulink; the obtained results are therefore qualitatively valid also for plants with very different characteristics and sizes. In facts physical models, even though to the detriment of the precision of the results, have a larger validity than the behavioral ones (“shadows” models), which are more employed by the constructors and derive from the experimental outputs obtained as answer to typical inputs.

The model is non-linear, because of the adiabatic
Among these inputs, only the following can be manipulated:

- the fuel flow (methane or oil), by means of a regulation servo-valve;
- the bypass valve, that shares the available heat between the two loads: the mechanical one at the rotor of the compressor-turbine-alternator unit, and the thermal one of the boiler for the water heating.

The other inputs are so considered like noises; the most important, as for amplitude and speed of action, are the requested electric power and the water flow to be heated.

**Outputs**
The model presents a lot of measurable outputs, some of an electromechanical nature (for example the speed of the rotor and the supplied electric energy), others of thermal kind (the air and gas flows, the compression ratio and the temperatures of gas).

It’s worth mentioning that the temperature of the gas delivered by the combustion chamber, both for its high value and the lack of space for the measurement, can’t be physically taken, but the model can on–line estimate it. Its value represents the main constraint restricting the output powers, considering his effects on the lifetime of blades.

The controlled variables, for which a set-point is imposed, are the following:

- the rotation speed of the shaft, from which a lot of system performances depend (first of all the air-gas flow and the mentioned gas temperature). Thanks to the converter, the rotation speed is not constrained to the electric frequency, so it can vary as one likes; in particular, it can be maintained as close as possible to the value corresponding to the maximum system efficiency (which in its turn depends on the required electric power), on condition that the combusted gas temperature does not exceed the limit imposed by the constructor;
- the temperature of the water provided to the boiler, that must vary as little as possible in comparison with the required value, to avoid big noises to the thermal users.

The other outputs are on-line available to verify the values of variables on which constructive, security or running constraints are fixed.

**Steady state performances**
The model allows the simulation of the plant in a large range of steady state conditions (for example varying the thermal/electric power ratio by means of the bypass valve placed in the recuperator, or the requested electric power), in order to analyze the values of the main internal variables (like the temperatures along the air-gas path) and deduce operational criteria which allow the respect of the technical limits.

**Dynamic performances**
The model simulates the dynamic behavior of the system, due to:

- the store of mechanic energy in the compressor-turbine-alternator unit: because of its low inertia, the rotation speed is very sensitive to imbalances between...
the driving torque provided by the turbine and the resistant torque of compressor and alternator; the store of air into the volume of the recuperator, air side; the store of heat into the metals of recuperator; the store of heat into the metals and the water contained in the boiler. Moreover the time required for the fuel combustion (fractions of a second) is considered; this time is very shorter than the other thermal time constants, but it’s significant to analyze the response of the plant in case of large and fast variations of the required electric power in islanded operation. For this purpose, tests with rotation speed regulated by means of the fuel flow are generally useful. The model use concentrated parameters, so it’s adequate to describe the system behavior only at low frequencies (corresponding to time constants of some seconds and more), useful to investigate the regulation systems and the main mechanical and thermal phenomena. The model is on the contrary inadequate to analyze fast phenomena like high frequency electric transients; thanks to the very quick response of the inverter, the electric part has been always assumed in steady state, i.e. the provided power coincides with the requested one, without taking into account the corresponding dynamic.

THE REGULATING SYSTEM

The process presents variable parameters and a strong interaction among its components; the regulating system is multivariable, with two inputs and two outputs; for its design the principle of non-interaction could be used, so to avoid that requests of thermal power influence the mechanical one and then the shaft speed and vice versa. The two regulating systems have nevertheless very different requirements. The regulation of the water temperature has few constraints, thanks to the thermal capacity of the boiler and to the big store represented by the distribution plants of hot water and heating, in comparison with the acceptable fluctuations of the hot water temperature; for example, in case of urban users this temperature can amply vary, up to a temporary interruption of its supply in extreme cases during critical contingencies, on condition that his average value keeps right on a long term basis. On the contrary, the regulation of the rotor speed has very stricter constraints. Therefore a frequency separation between the two regulation systems is possible: a quick control is carried out on the rotation speed by means of the fuel flow, while a slower control system (of an integral nature) controls the temperature of the hot water by acting on the bypass valve. To avoid excessive values of the gas temperature at the combustion chamber exit, in the big gas turbines the air flow is partialized by means of IGV valves; likewise, in the case of microturbines the same result is obtained by varying the set point of rotation speed and the position of the bypass valve. Then the regulating scheme assumes the typical “cascade” framework: two external loops for the limitation of the mentioned gas temperature, plus an internal one that acts on the fuel flow for the speed regulation. When the limit temperature is exceeded, the first external loop modifies with a dynamic function the optimal value of the speed set point (programmed for the maximum efficiency), while the second loop transitarily acts on the position of the bypass valve (whose steady state value is controlled by the regulator of the hot water temperature). Because of the slow action of these regulations, a fast limiting device, that promptly acts on the fuel flow in case of a dangerous exceeding of the above mentioned limit temperature, is installed. This scheme assures a safe behavior against failures or unexpected restrictions of the fuel supply, using the stores of the system up to the limit allowed by the performance of the machinery. To avoid that large variations of the required electric power can generate excessive speed variations of the shaft or unacceptable unbalances of electrical vs thermal power demand, a load demand programmer is also superimposed, with open chain restrictions to the admitted range of variations.

SIMULATIONS AND RESULTS

The steady state simulations carried out with the model confirm that the current technology makes this kind of plants suitable for dispersed cogeneration when the electric/thermal power ratio, whose modulation is allowed by the bypass valve, is in the 2-5 range, as already declared by the microturbines constructors (1). These values seem to be compatible (4) with the requirements of four main categories of users:

- industrial loads with discontinuous running;
- industrial loads with continuous running;
- tertiary winter loads;
- tertiary summer loads (assuming the use of chiller systems for the air conditioning).

As example of the dynamic simulations the model allows, in Fig. 3 the responses to a +10% step variation of the requested electric power (from 45 to 50 kW) are shown. This simulation demonstrates the very prompt response of the mechanical power provided at the shaft, obtained by modulating the fuel flow, so step variations of the required electric power are allowed without unacceptable perturbations to the main operating variables.

1 The rotation speed is the variable interacting with nearly all the others in the system, because it establishes the flow of compressed air, which in its turn influences the turbine gas inlet temperature, thence the temperatures at recuperator and boiler. For example, an increase in fuel flow implies initially the growth of temperatures and therefore of the power delivered by the turbine; but the consequent rise of the rotation speed increases the air and gas flow, so that the output temperatures of the combustion chamber and of the turbine go down to a lower value than the initial one.
In this example, a 10% variation of the electric power, with the corresponding modifications of fuel flow (without overfiring) and rotation speed set point, entails a fluctuation of the gas temperature at turbine exit (the most restricting point) within 5%.

CONCLUSIONS

In another paper (5), the usefulness of employing also diffused microgeneration to supply the ancillary services, which the electric system needs to operate according to quality and security requirements, is discussed; some of these services involve important variations of the provided active power, e.g. primary frequency regulation with perturbed or islanded interconnected grid, capability to load rejection and black start.

The described model allows the valulation of the solutions offered by co-generation systems with microturbines, in order to identify which problems can affect this kind of plants and design the regulating and protection system.

The simulations carried out with this model show that the problem concerning the supply of heat to the local users can be temporarily separated from the quick and large electric power variations required to support the main grid in case of important perturbations, or the local islanded grid in case of large out-of-services with separation from the main network; thanks to the current technology, these power demands can be met without exceeding the limitations posed by the constructor as regards to the rotation speed and the gas temperatures.

These considerations, together with the quick start of turbines and the prompt power and voltage modulation allowed by inverters, make systems with microturbine very useful to contribute to the different needs of an electric grid.

References

1. Bowman Power Systems Ltd, Technical documentation for cogeneration microturbines, Ocean Quan, Belvidere Road, Southampton SO14 5QY UK.
Appendix: The mathematical model

### Compressor

**Inputs:**
- exit pressure $P_e$ [bar]
- rotation speed $\omega$ [kprm]
- external air temperature $T_a$ [°C]
- external air pressure $P_a$ [bar]

**Outputs:**
- compressed air flow $q_a$ [kg/s]
- reduced gas flow $q_{fr}$ [kg/s]
- temperature $T_f$ [°C]

$$q_a = \rho_c \cdot n_i \cdot \Phi \cdot \varnothing$$

where:

$$\Phi = \Phi_0 \cdot f_r(t_c) \text{ (characteristic flow number)} \quad (*)$$

$$r_c = \frac{P_{sc}}{P_e} \text{ (compression ratio)}$$

$$T_{ci} = T_a \cdot \left(1 + \frac{\gamma - 1}{\gamma} \right) r_c^{(\gamma-1)/\gamma} - 1$$

where:

$$\gamma = c_p / c_v, \quad \zeta = \zeta_0 \cdot f_c(t_r) \text{ (efficiency''') \quad (*)}$$

$$P_c = q_c \cdot c_p \cdot (T_{ci} - T_o)$$

### Combustor

**Inputs:**
- compressed air flow $q_a$ [kg/s]
- fuel flow $q_f$ [kg/s]
- heated air temperature $T_{ha}$ [°C]

**Outputs:**
- gas flow $q_g$ [kg/s]
- combusted gas temperature $T_c$ [°C]

$$q_f = q_a + q_{cb}, \quad T_f = \frac{q_c \cdot c_p \cdot T_{ci} + q_{ca} \cdot c_{ca} + q_{am} \cdot P_m}{q_f \cdot \varnothing}$$

### Recuperator

**Inputs:**
- gas bypass valve opening $K_b$ (p.u.)
- exit water flow $q_w$ [kg/s]
- external air temperature $T_a$ [°C]
- exit gas temperature $T_{ga}$ [°C]

**Outputs:**
- reduced gas flow $q_{fr}$ [kg/s]
- heated air temperature $T_{ha}$ [°C]
- average metal temperature $T_m$ [°C]
- gas to metal heat flow $Q$ [kcal/s]
- metal to air heat flow $Q_a$ [kcal/s]

$$q_{fr} = q_f \cdot (1 - K_b) \cdot q_f - Q_m = M_m \cdot c_m \cdot \frac{dT_m}{dt}$$

$$Q_f = q_f \cdot c_f \cdot (T_{fr} - T_{ha}) = S_m \cdot \Gamma_m \cdot \frac{T_{fr} + T_{ha} - T_m}{2}$$

$$Q_c = q_c \cdot c_p \cdot (T_{cu} - T_{ci}) = S_m \cdot \Gamma_m \cdot \frac{T_{cu} + T_{ci} - T_m}{2}$$

### Mixer

**Inputs:**
- exit gas flow $q_a$ [kg/s]
- reduced gas flow $q_{fr}$ [kg/s]
- recuperator exit gas temperature $T_{fu}$ [°C]
- turbine exit gas temperature $T_{g}$ [°C]

**Outputs:**
- mixed gas temperature $T_m$ [°C]

$$T_m = \frac{T_{fu} \cdot q_f + T_g \cdot (q_{fr} - q_f)}{q_f}$$

### Turbine

**Inputs:**
- combusted gas temperature $T_{fu}$ [°C]
- gas flow $q_f$ [kg/s]
- exit gas temperature $T_{fr}$ [°C]
- mechanical power $P_r$ [kcal/s]

**Outputs:**
- pressure at turbine entrance $P_{sc}$ [bar]
- exit gas flow $q_{fr}$ [kg/s]
- exit gas temperature $T_{fr}$ [°C]

$$V \frac{dP_{sc}}{dt} = q_f - q_{fr}$$

$$q_f = K_r \cdot \frac{P_{sc}}{T_{sc}} \cdot T_{fr} = T_{fr} \cdot \left(1 + \frac{1}{5} \left(\frac{P_{sc}}{T_{sc}}\right)^{\frac{1}{1.1}} - 1\right) \cdot \gamma_f = \frac{c_f}{c_v}$$

$$P_r = q_f \cdot c_p \cdot (T_f - T_{sc})$$

### Boiler

**Inputs:**
- water flow $q_w$ [kg/s]
- entry water temperature $T_e$ [°C]
- exit gas flow $q_{fr}$ [kg/s]
- mixed gas temperature $T_{ga}$ [°C]

**Outputs:**
- exhausted gas temperature $T_{ga}$ [°C]
- heated water temperature $T_{ha}$ [°C]
- thermal power $P_e$ [kcal/s]

$$Q_a + q_u \cdot c_u \cdot (T_m - T_h) = (M_m \cdot c_m + M_w \cdot c_w) \cdot \frac{dT_w}{dt}$$

$$Q_u = q_f \cdot c_f \cdot (T_{fr} - T_f)$$

$$Q_u = S_u \cdot \Gamma_{u} \cdot \left(\frac{T_{pu} + T_{ex}}{2} - T_u\right) \cdot T_u = \frac{T_{pu} + T_{ex}}{2}$$

### Compressor-turbine shaft

**Inputs:**
- mechanical power of turbine $P_r$ [kcal/s]
- mechanical power of compressor $P_c$ [kcal/s]
- requested electric power $P_e$ [kW]

**Outputs:**
- rotation speed $\omega$ [kprm]

$$\frac{(P_c - P_e)}{A \cdot \varnothing} - C_a = J \frac{d\varnothing}{dt} \quad C_a = C_0 + C_1 \cdot \varnothing$$

### Parameters

- $\rho_c$: cold air density
- $n_i$: compressor dimension
- $\Phi_0$: compressor base flow number
- $\zeta_0$: air compressor base efficiency
- $c_{ca}$: fuel specific heat
- $c_{am}$: gas specific heats
- $V$: recuperator volume, air side
- $\delta P_r$: dependence of gas density from pressure (T cost)
- $K_r$: turbine equivalent admittance (Stodola coeff.)
- $\eta_r$: turbine base efficiency
- $S_m$, $S_{cm}$: exchange surfaces of recuperator (gas and air side)
- $\Gamma_m$, $\Gamma_{m}$: exchange coeff. of recuperator
- $M_m$, $c_m$: mass and specific heat of metal in recuperator
- $S_a$: boiler exchange surface (gas side)
- $\Gamma_a$: boiler exchange coeff.
- $M_w$, $c_w$: mass and specific heat of metal in boiler
- $C_0$, $C_1$: friction factor at shaft
- $A$: 4.186 kWh/kcal/s
- $J$: inertia of compressor-turbine-alternator unit

(*) $f_r(t_r)$ and $f_c(t_r)$ will be defined by means of experimental tests; at present their value has been assumed 1.